Fundamentals of Gear Lubrication

for

Department of the Navy

Bureau of Aeronautics

Contract No. 53-356-c

Amendment No. 14

Final Report

June, 1956 - June, 1957

SHELL DEVELOPMENT COMPANY
EMERYVILLE, CALIFORNIA

FUNDAMENTALS OF GEAR LUBRICATION

for

Department of the Navy
Bureau of Aeronautics

under

CONTRACT NO. 53-356-c - AMENDMENT NO. 14

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SHELL DEVELOPMENT COMPANY
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ABSTRACT

This Final Summary Report describes the work performed during the fourth year of operation under Contract No. 53-356c , covering the period from June, 1956 to June, 1957. The subjects investigated were:

1. The effect of viscosity on load carrying capacities of oils. Here, the data on viscosity - pressure - temperature characteristics of the sixteen oils used throughout these investigations are presented and discussed. The studies of relations between load carrying capacities of oils and their viscosities are described. A theoretical approach used in these studies was especially fruitful and is presented in detail.

2. The studies of the effect of hardness on gear performance are described. These studies showed numerous advantages of hard gears over soft ones in power transmission applications.

3. The effect of length of contact on load carrying capacity is described. It was shown that this effect is twofold: The total load carried by gears increases with the increase in contact length, while unit load decreases.

4. Scoring performances of oils in the speed range between 20,000 and 30,000 rpm are described. A substantial increase in load carrying capacities of oils at these high speeds was noted.

5. Performances of gears and gear lubricants at high temperatures between 400°F and 600°F are described. No unusual or new gear failures were observed. Load carrying capacities of oils, as a rule, decreased with the increase in temperature; but at 400°F or higher, lubricants formed gummy deposits which act as protective coatings and improve the scoring performance of gears.

6. The expanded data on the effect of speed on gear wear are presented. Previous findings that load carrying capacities of oils are inversely related to wear were confirmed.

7. Scoring performances of Wright Air Development Center oils GTO-38 and WADC reference oil "B" are described.

8. The studies of extreme pressure lubrication are presented. These include the studies of the mechanism of formation and action of extreme pressure films, studies of phosphorus as an extreme pressure agent, and studies of the effect of concentration and characteristics of di-additive systems.

The report is concluded by a brief review of the work performed during the entire four year period and final theoretical and practical conclusions are formulated.

FUNDAMENTALS OF GEAR LUBRICATION

Introduction

The work performed under the Bureau of Aeronautics Contract No. 53-356c Amendments 12 and 14 during the period from June, 1956 to June, 1957 is presented in a summarized form. In addition the results obtained throughout the four years of the investigations under this Contract are briefly reviewed, and theoretical and practical conclusions derived from these investigations are presented.

The work performed during the fourth year of these investigations consisted of the studies of relations between viscosities and load carrying capacities of oils, studies of the effects of high speeds and high temperatures on gear lubrication, expanded studies of extreme pressure lubrication and studies of the effects of hardness and length of contact. In addition scoring performances of two Wright Air Development Center reference oils were investigated, and the data on the effect of speed on gear wear were expanded.

The Effect of Viscosity on Load Carrying Capacity of Oils

Viscosity-Pressure-Temperature Characteristics of Oils

Introduction. In our studies of the mechanism of gear lubrication the effects of various operating variables and gear geometry factors were investigated using a large number of mineral, synthetic and extreme pressure compounded oils. The results of these tests showed that the heavier was the lubricant the higher was its load carrying capacity and the smaller the wear. Thus, when a series of mineral oils ranging in viscosity from 9.97 cs to 393.0 cs at 100°F was tested for load carrying capacity at various speeds (see Table 8 of Summary Report S-13605), the oils lined up according to their viscosity. Similar relations were noted with synthetic oils such as silicones and polyglycols (see Table 8, 9 and 10 of Summary Report S-13649). However, comparison of the performances of mineral and synthetic oils showed that some of the synthetic oils exhibited higher load carrying capacities than did the much heavier mineral oils and that among themselves the synthetic oils of similar viscosity differed widely. This fact was taken as an indication that the viscosity of oils, as measured at atmospheric pressure and some arbitrarily chosen temperature, such as 100°F or 210°F, cannot be used as a measure of the performance characteristic of lubricants. It was more logical to assume that viscosities of the oil films under contact conditions would be a better criterion.

As a first step to widen our understanding of this phenomenon, it was decided to establish the relations between the viscosities of the lubricants used and such variables as pressure, temperature and shear rate. It was suspected that differences in these characteristics could explain the differences in performances of lubricants. Knowing the viscosity - pressure - temperature characteristics of oils it appeared also possible to estimate the viscosity within the contacting films and to check if a relation exists between these viscosities and load carrying capacities.

For the purpose of establishing the viscosity - pressure - temperature characteristics of oils, a high pressure viscometer was constructed (see page 10 of Summary Report S-13694), and the viscosities of sixteen oils used throughout these investigations were measured at pressures varying from atmospheric to 120,000 psi and temperatures of 25°C, 100°C and 150°C. The results of these measurements are presented in Table 1 and Table 2 and graphically in Figures 1 to 14.

Discussion of Results. The limiting factors in these tests were: (1) a viscosity of 1000 poises, (2) solidification of the test fluid, (3) pressure limitation of the equipment (120,000 psi), and (4) temperature limitation (150°C). Within these limits, relatively few difficulties were experienced throughout this work, and the data obtained are considered as accurate as that of the American Society of Mechanical Engineers sponsored project at Harvard University. As an example, comparisons of viscosity data of Plexos 201 obtained at Harvard and Shell Development are given in Figure 1.

Viscosities of oils tested at three temperatures are plotted as a function of pressure in Figures 2 to 7. It is of interest that the log viscosity - pressure relations of uncompounded mineral oils at the lowest temperature used (25°C) are practically linear and at elevated temperatures (100°C and 150°C) these lines begin to curve. For the synthetic oils the log viscosity - pressure relations are represented by somewhat curved lines, even at 25°C. Comparison of the viscosity - pressure curves for different lubricants shows that the effect of pressure on viscosity is larger for mineral oils than for synthetic oils. For each series of lubricants this effect is larger the higher the nominal viscosity of the lubricant.

Typical charts for the effect of temperature on viscosity at different pressures are given in Figures 8 to 11. Figure 12 is a summary chart for the data. Here the relations between viscosity and absolute temperature have been represented by straight lines on a log - log chart, which is considered permissible for "practical" or "engineering" uses.

With most of the oils used in this research the pressure within the lubricating film between meshing gear teeth was much higher than that used in determining their viscosity - pressure characteristics. Hence, extrapolation of the data is necessary to obtain estimates of viscosities at gear operating conditions. To facilitate such an extrapolation, several forms of empirical equations have been tried to fit the data. Good results have been obtained using an empirical equation of the form:

$$\log \eta = A + \frac{B(P + C)}{T^3}$$

Where η = viscosity (absolute mksu)
P = absolute pressure in Newtons/m^2
T = absolute temperature in ^{o}K
A, B, C = constants.

The values of the constants for each oil pertaining to this equation were calculated and are given in Table 2. From the comparison of the constants for different oils it appears that the constant "B" is related to the chemical nature of the lubricant. Several examples of correlation between measured and calculated values for viscosity are given in Figure 13.

Figures 1 through 13 follow

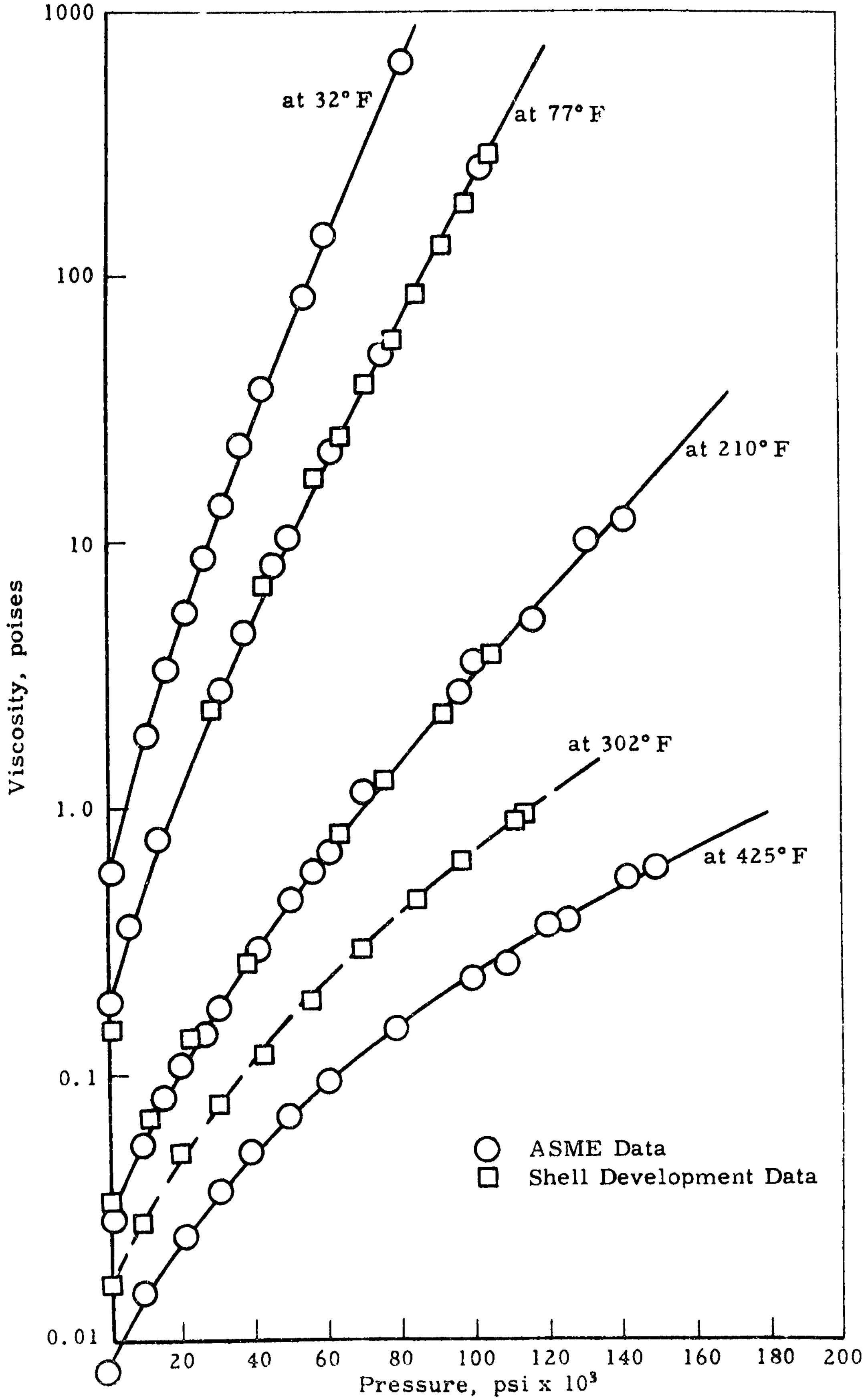

Figure 1. COMPARISON OF ASME AND SHELL DEVELOPMENT DATA

Viscosity - Pressure Relation for Plexol 201

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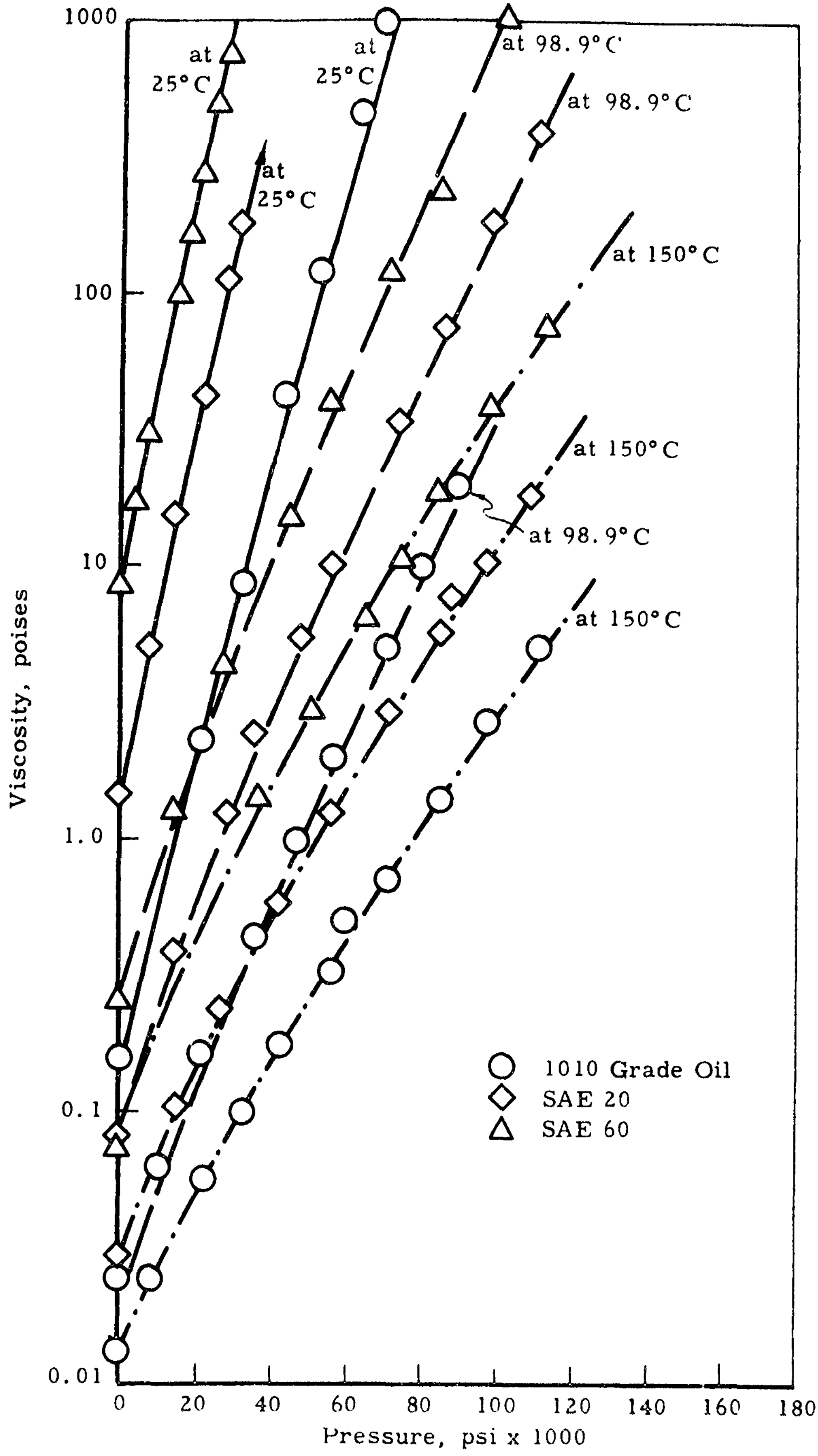

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Figure 2. PRESSURE - VISCOSITY RELATION FOR MINERAL OILS

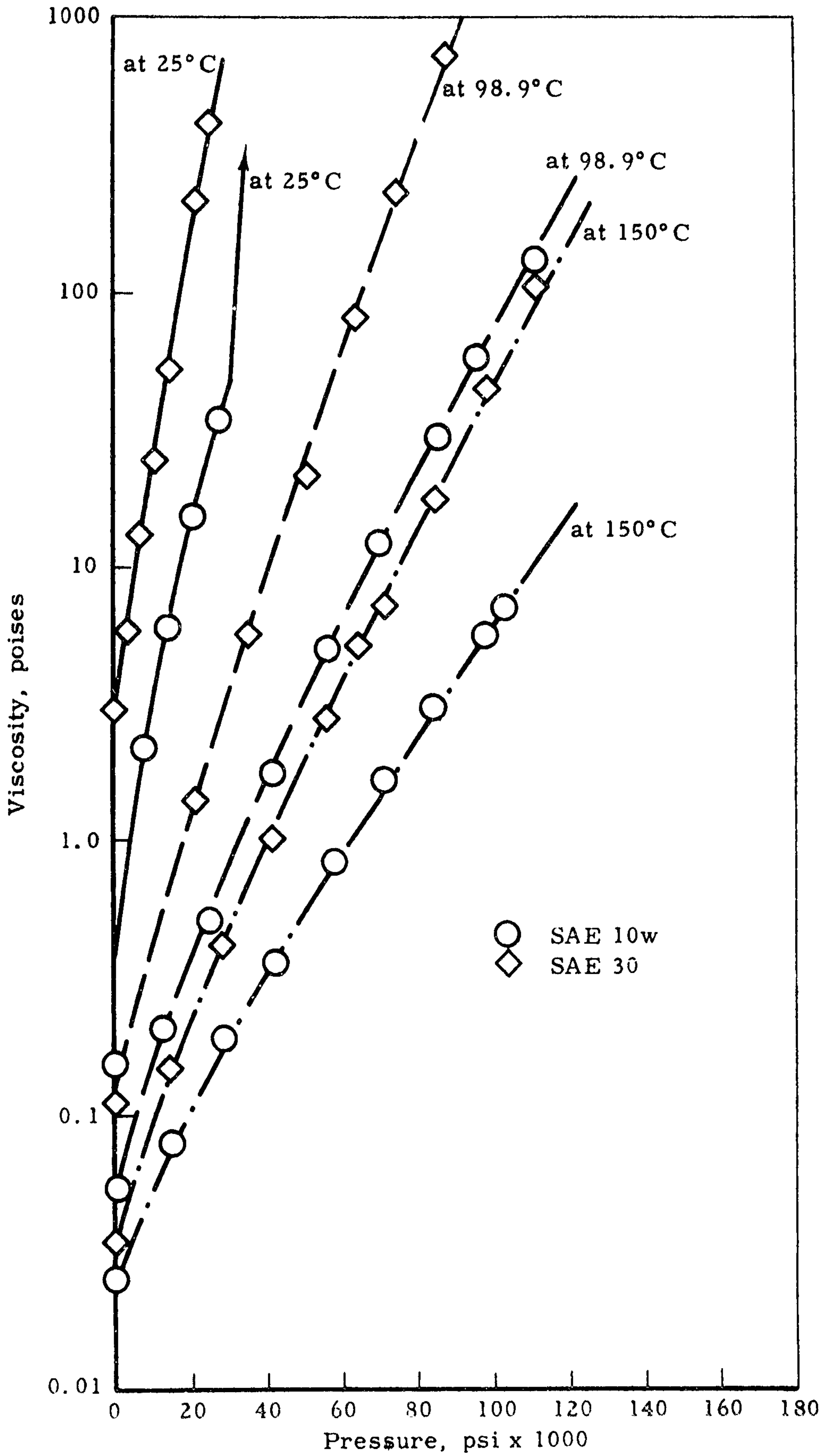

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Figure 3. PRESSURE - VISCOSITY RELATION FOR MINERAL OILS

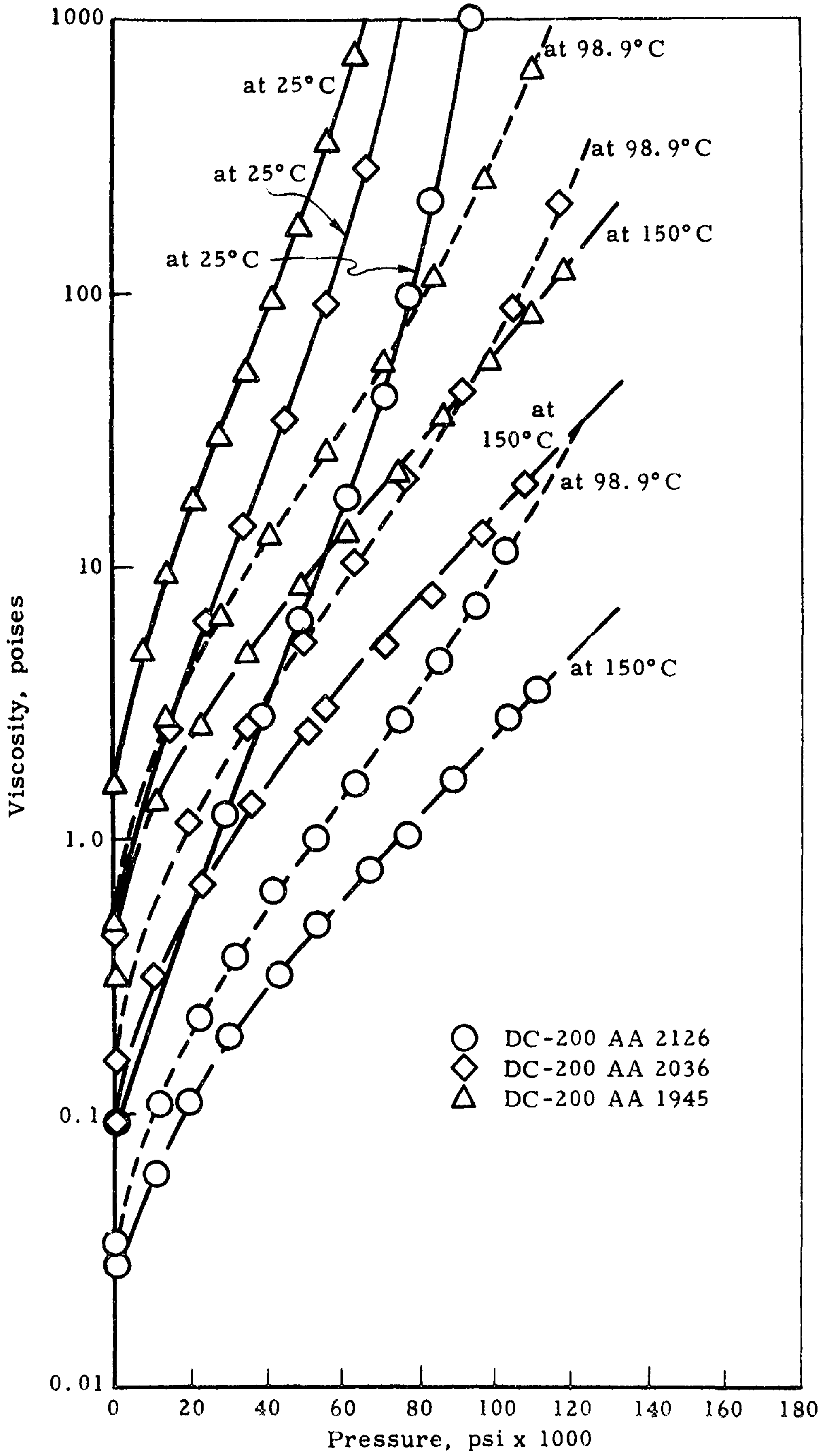

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Figure 4. PRESSURE - VISCOSITY RELATION FOR SILICONES (DC-200 SERIES)

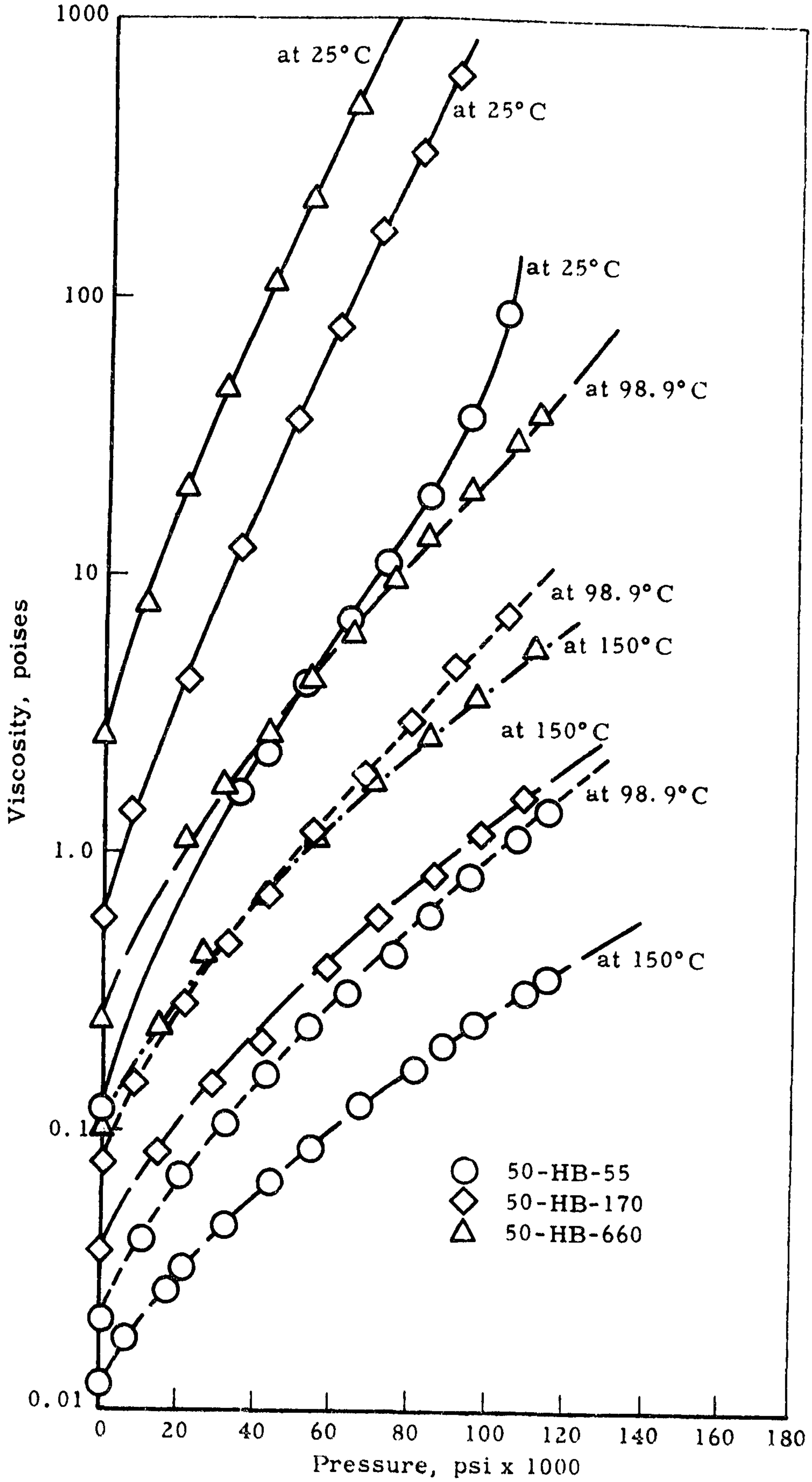

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Figure 5. PRESSURE VISCOSITY RELATION FOR UCON-50-HB SERIES

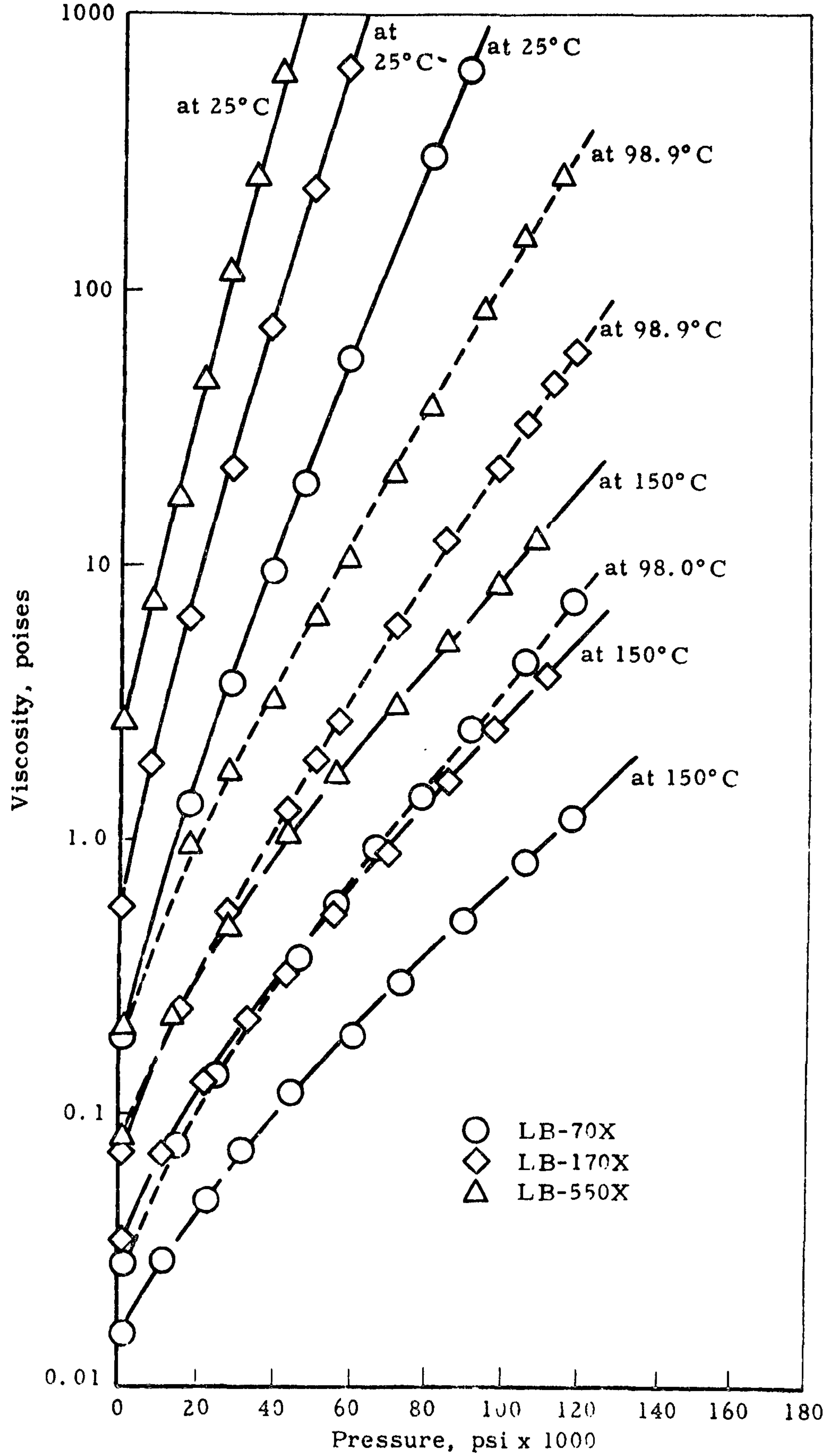

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Figure 6. PRESSURE - VISCOSITY RELATION FOR UCON-LB SERIES

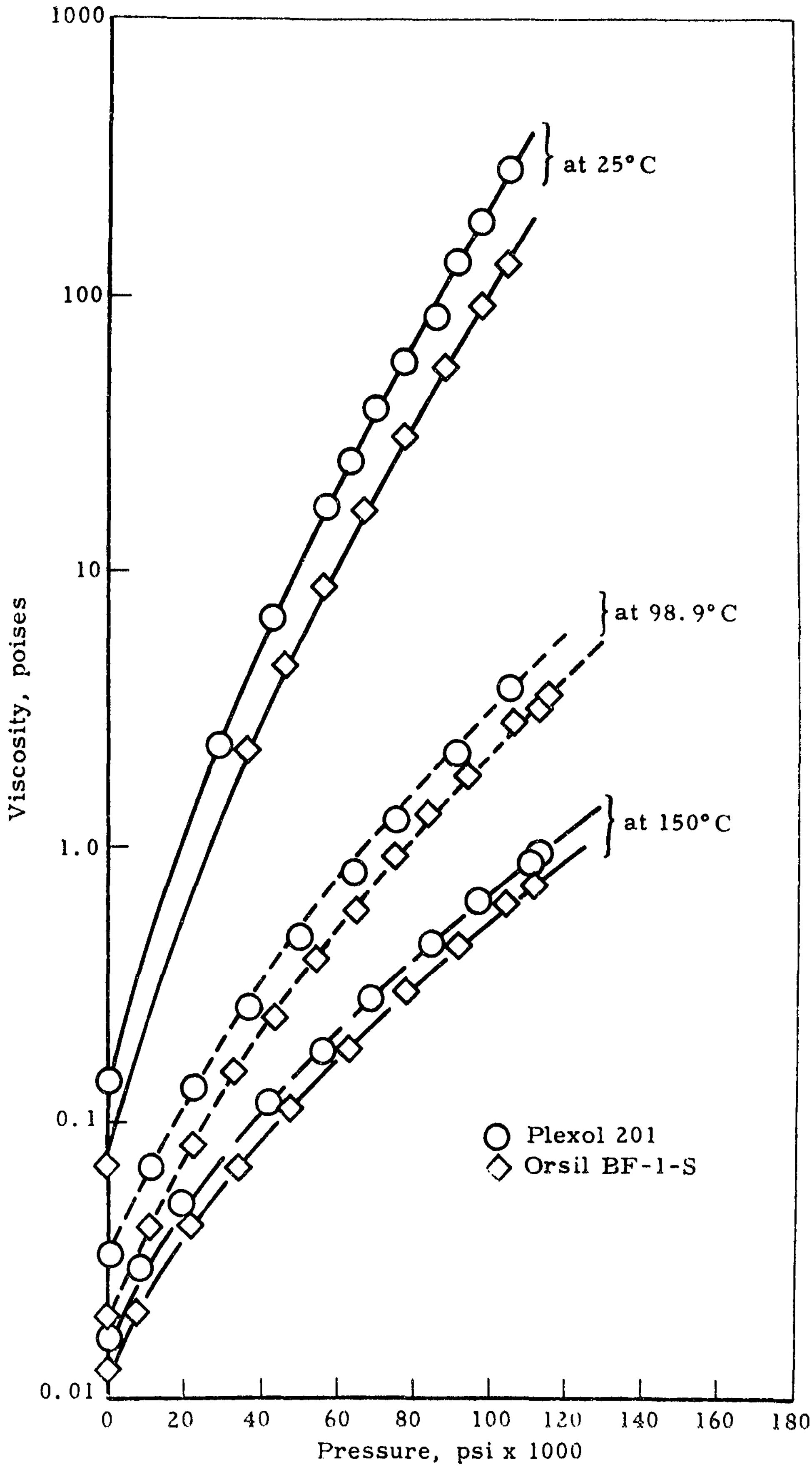

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Figure 7. PRESSURE - VISCOSITY RELATION FOR DI-ESTER AND SILICATE ESTER

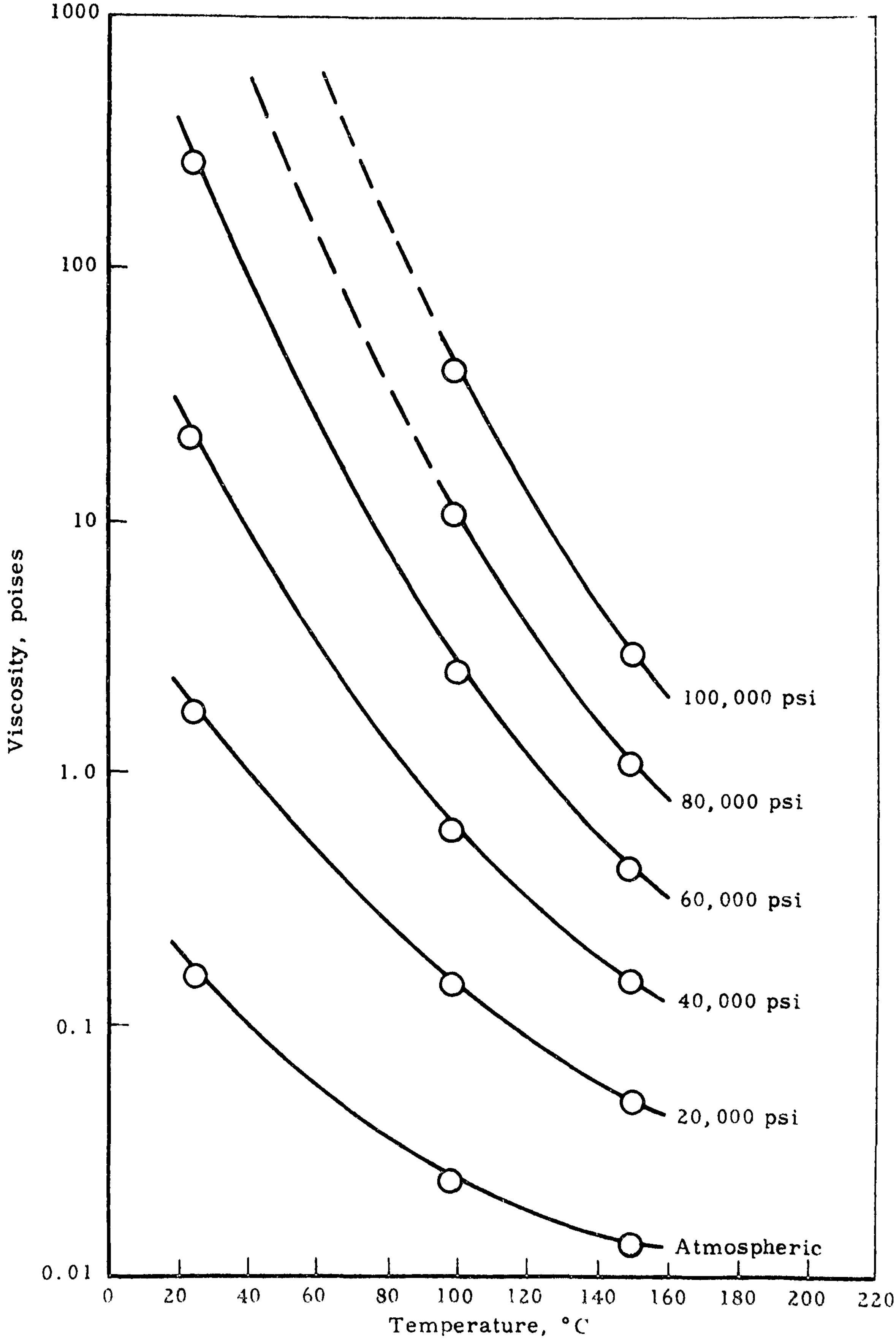

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Figure 8. TEMPERATURE - VISCOSITY RELATION FOR 1010 MINERAL OIL

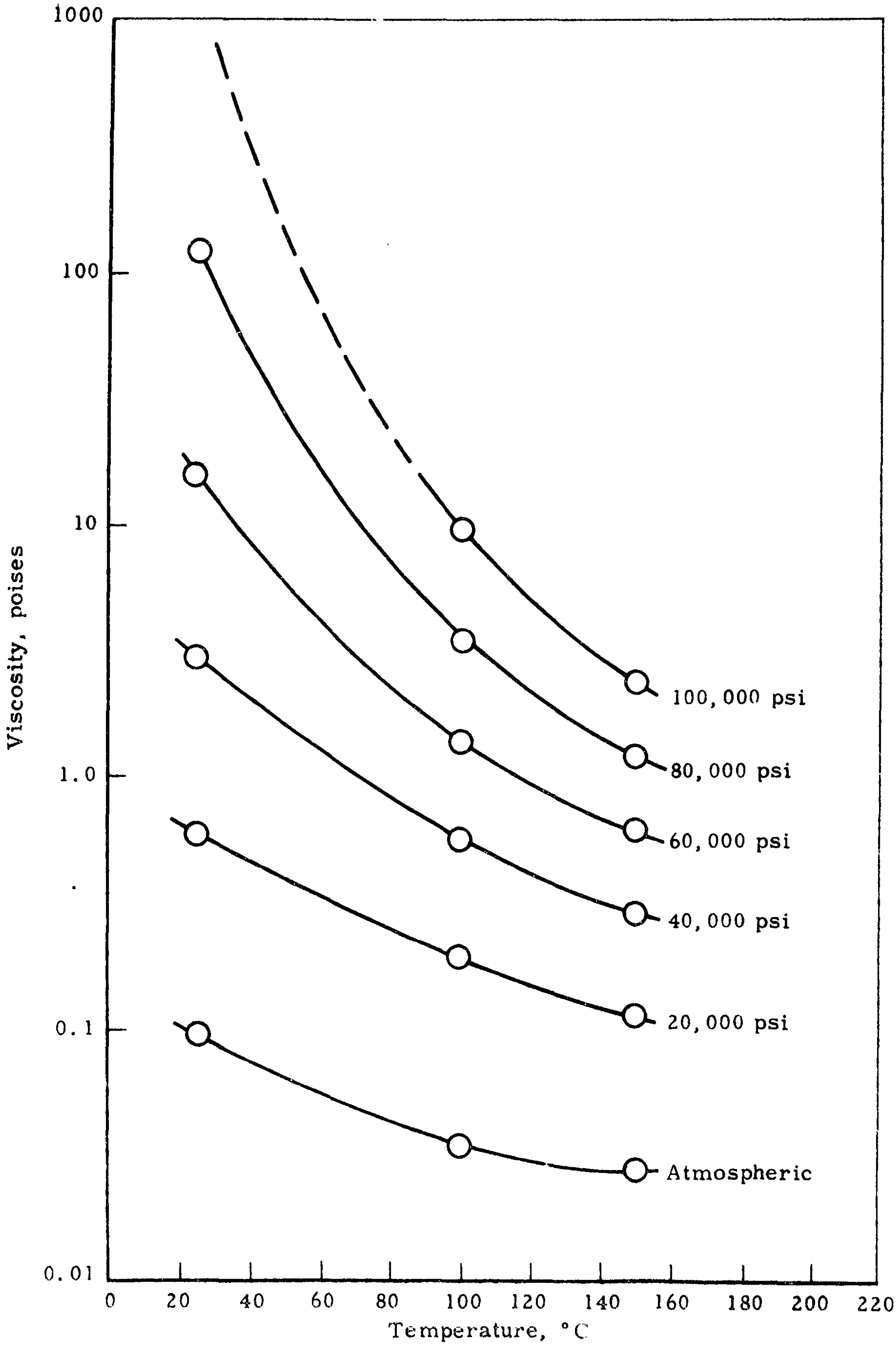

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Figure 9. TEMPERATURE - VISCOSITY RELATION FOR SILICONE DC-200 AA 2126

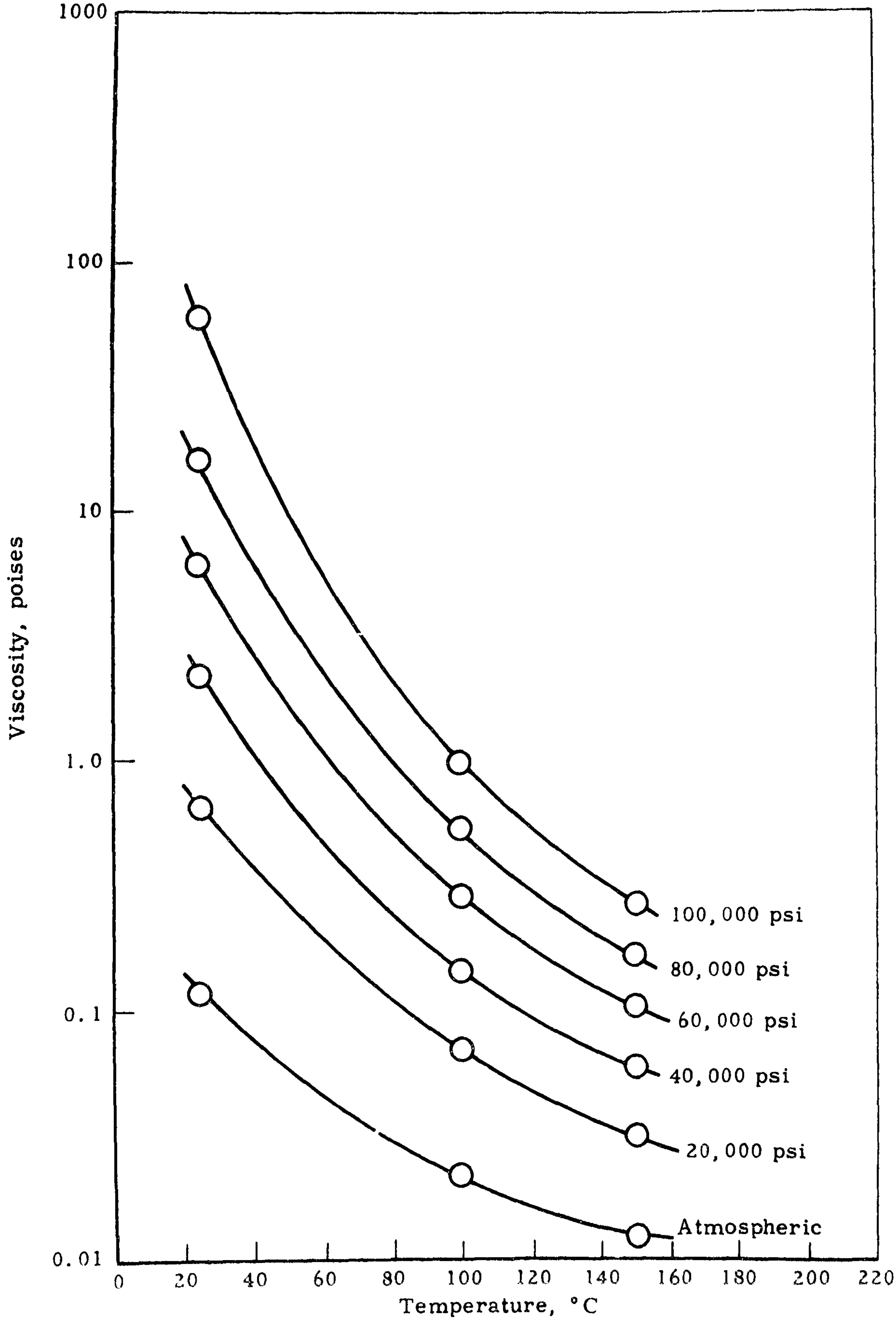

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Figure 10. TEMPERATURE - VISCOSITY RELATION FOR UCON 50-HB-55

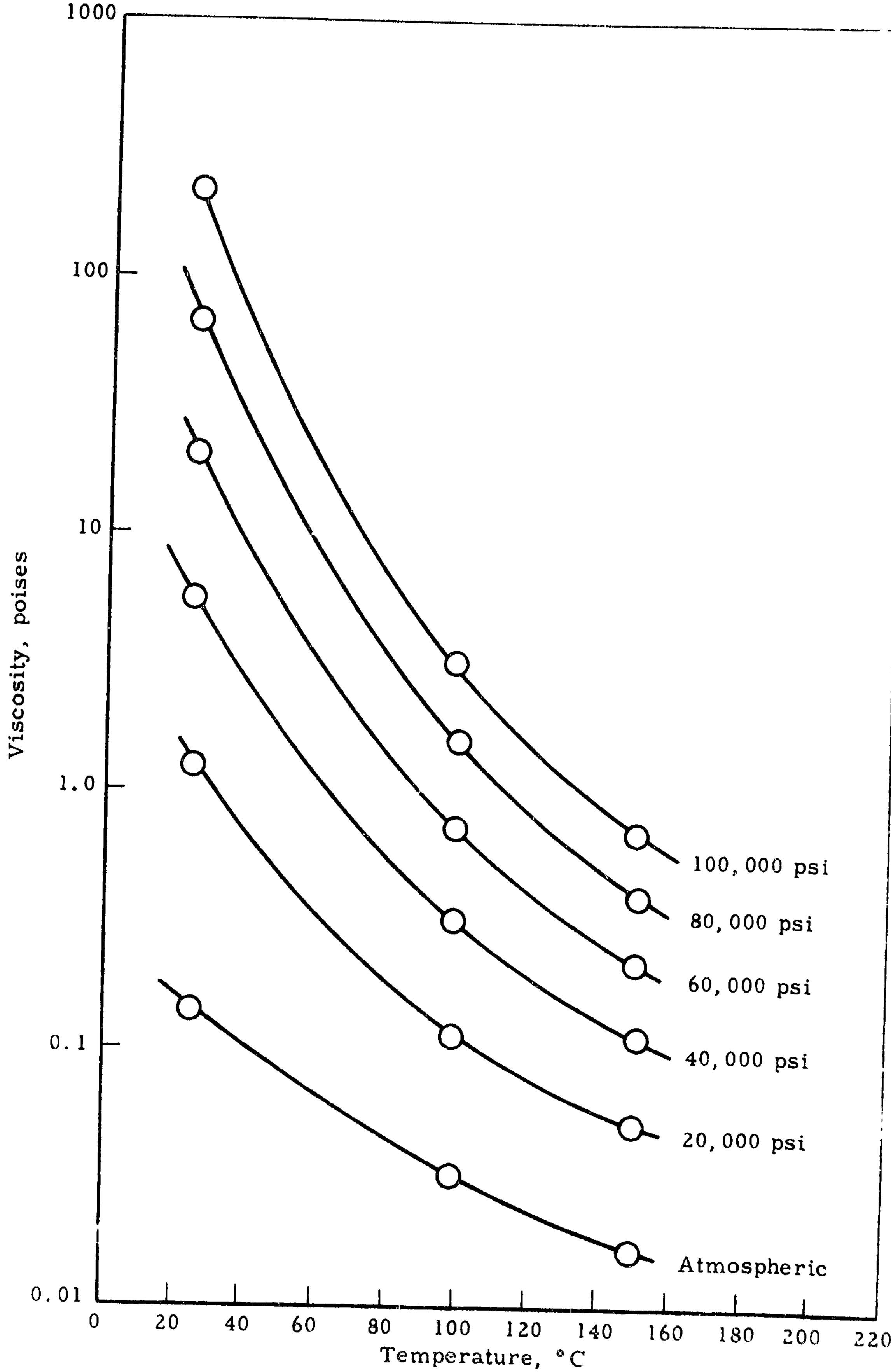

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Figure 11. TEMPERATURE - VISCOSITY RELATION FOR PLEXOL 201

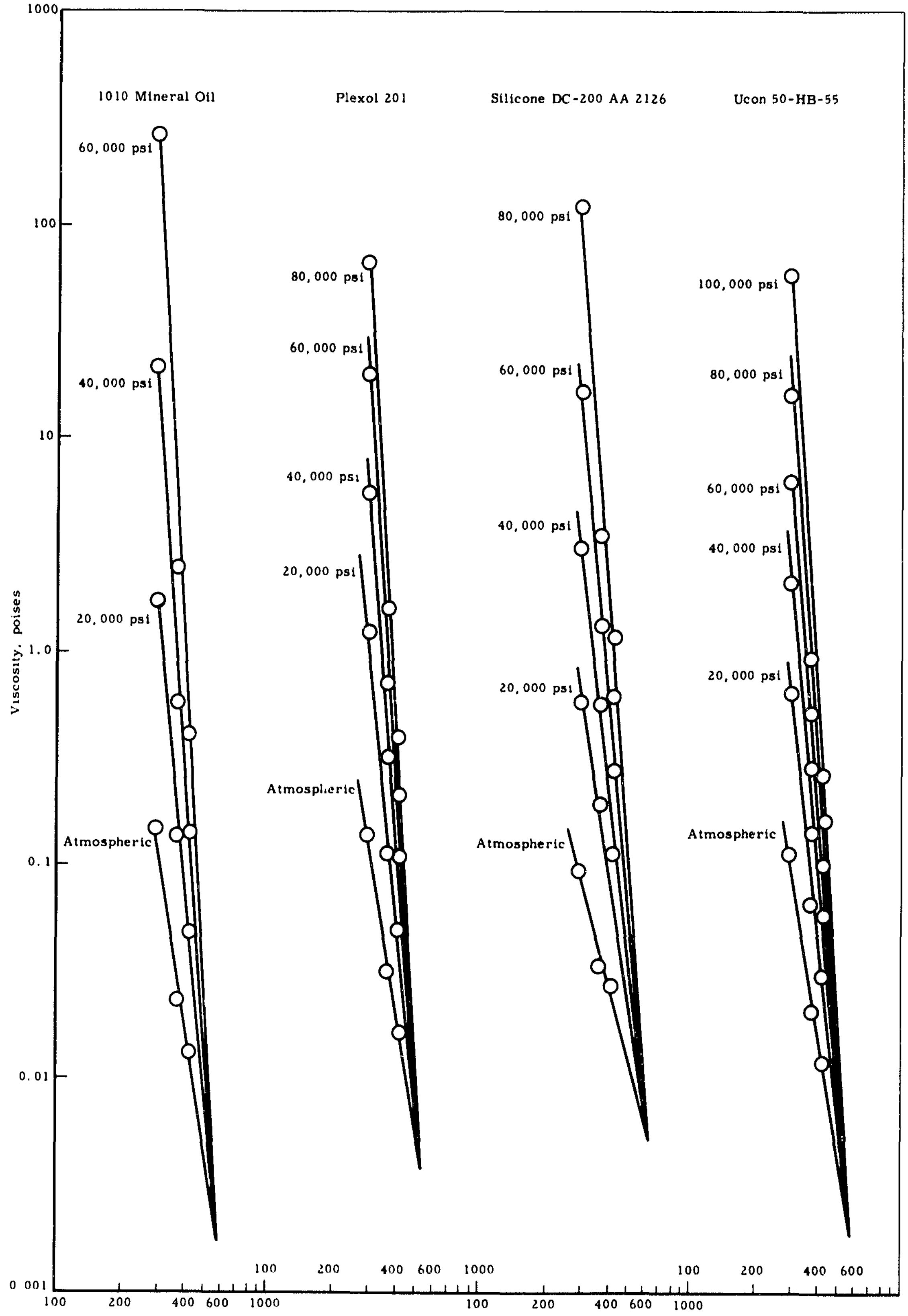

Figure 12. VISCOSITY - TEMPERATURE RELATION

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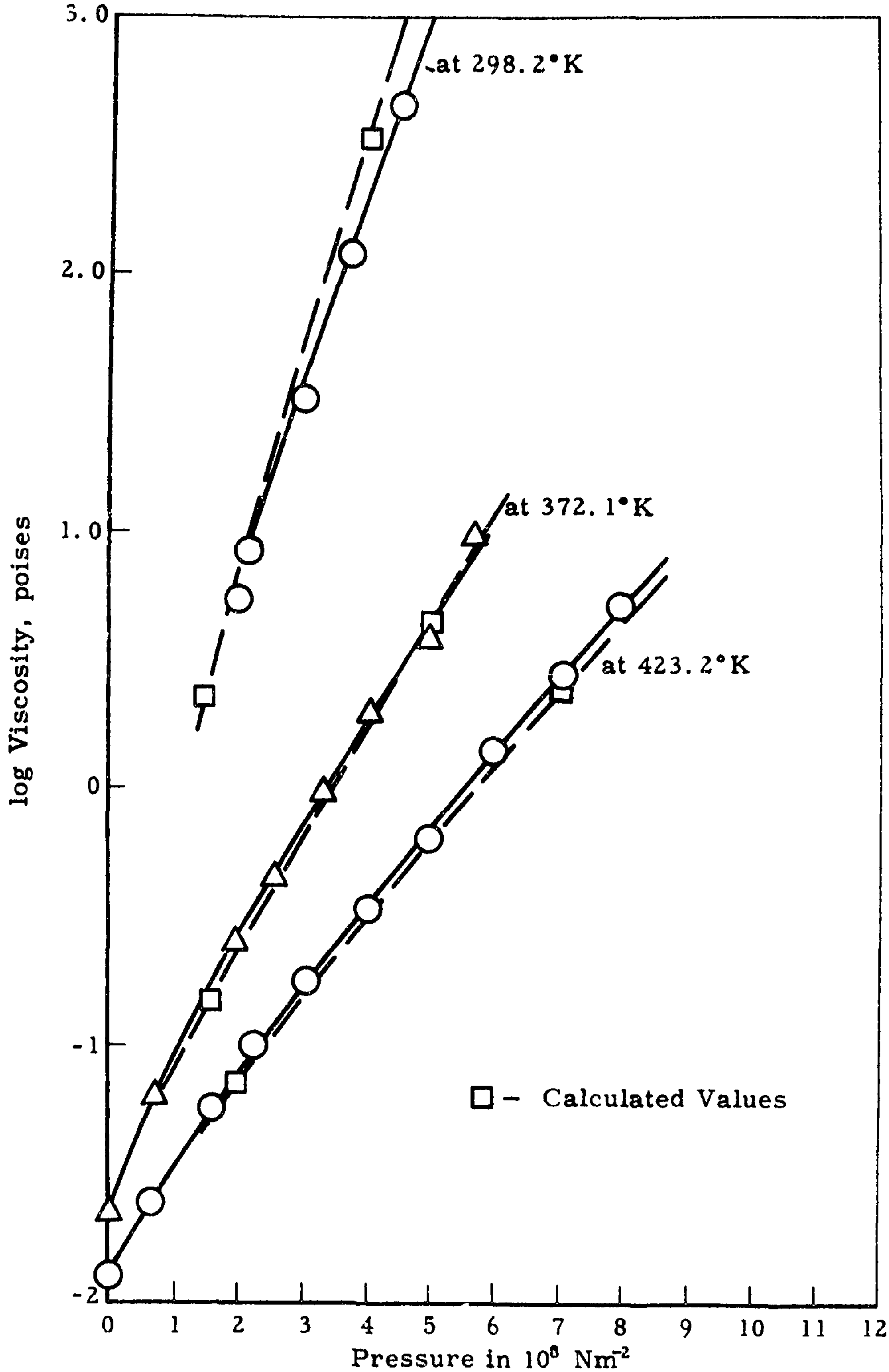

Figure 13. CORRELATION BETWEEN CALCULATED AND EXPERIMENTAL VALUES OF VISCOSITY OF 1010 GRADE MINERAL OIL

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Table 1. VISCOSITY-PRESSURE-TEMPERATURE RELATION

Mineral Oils

1010 Grade Oil

25°C		98.9°C		150°C	
psi	Poises	psi	Poises	psi	Poises
0	.152	0	.023	0	.013
21,300	2.24	1[illegible],500	.061	9,200	.024
31,300	8.26	23,100	.159	23,000	.057
42,000	32.7	36,300	.431	32,400	.097
52,500	120.4	46,800	.952	43,300	.177
62,800	456.4	57,200	1.965	56,400	.328
69,400	1001.1	70,300	5.076	71.200	.716
		80,000	10.07	84,600	1.39
		89,500	19.87	97,900	2.69
				111,400	4.96

SAE 10 Mineral Oil

25°C		98.9°C		150°C	
psi	Poises	psi	Poises	psi	Poises
0	.152	0	.055	0	.025
7,200	2.25	13,000	.206	14,900	.078
14,100	5.89	24,500	.51	28,800	.182
20,600	15.4	41,600	1.76	42,300	.363
27,700	34.5	56,400	4.91	58,400	.833
35,000	Solid	70,300	12.0	71,700	1.63
		85,000	29.3	84,700	3.04
		96,000	56.3	98,100	5.52
		110,400	130.0	103,600	6.92

SAE 20 Mineral Oil

25°C		98.9°C		150°C	
psi	Poises	psi	Poises	psi	Poises
0	1.43	0	.0815	0	.030
7,700	4.79	14,100	.375	15,000	.106
14,400	15.2	28,100	1.22	27,100	.235
21,300	42.5	35,800	2.42	42,000	.591
28,000	114.3	47,400	5.48	56,000	1.29
31,100	175.0	56,000	10.2	71,000	2.91
33,000	Solid	73,000	33.5	84,000	5.71
		85,100	75.6	96,800	10.5
		98,000	190.4	108,500	17.9
		110,000	393.0		

SAE 30 Mineral Oil

25°C		98.9°C		150°C	
psi	Poises	psi	Poises	psi	Poises
0	3.01	0	.112	0	.035
3200	5.88	21,600	1.40	15,400	.149
7200	13.0	35,800	5.61	28,000	.408
10,200	24.7	50,200	21.5	41,300	1.01
14,100	52.7	63,900	78.9	55,300	2.76
21,300	216.9	74,900	222.5	64,600	5.14
24,500	419.4	86,700	706.2	70,800	7.27
		97,200	2005.0	84,000	17.5
				98,100	43.7
				112,600	102.5

SAE 60 Mineral Oil

25°C		98.9°C		150°C	
psi	Poises	psi	Poises	psi	Poises
0	8.22	0	.256	0	.078
3,600	16.8	13,600	1.23	36,400	1.39
7,200	29.9	27,000	4.32	50,200	3.00
14,100	97.2	44,900	15.2	64,200	6.53
17,600	169.8	55,100	40.9	73,500	10.7
20,600	273.9	70,500	122.1	84,100	18.9
24,500	492.7	84,000	336.7	98,700	38.4
27,200	758.9	93,700	664.1	113,500	76.7
		100,900	1098.0		
		103,200	1240.0		

(Continued)

Table 1 (Contd-1). VISCOSITY-PRESSURE-TEMPERATURE RELATION

Silicones

25°C		98.9°C		150°C	
psi	Poises	psi	Poises	psi	Poises
DC-200 AA-2126					
0	.095	0	.033	0	.028
28,600	1.42	11,200	.107	10,200	.060
38,800	2.83	21,800	.221	19,500	.109
49,000	6.21	31,600	.375	30,000	.192
61,300	17.9	42,000	.625	43,300	.328
70,500	42.1	52,500	1.01	53,600	.483
78,000	98.3	63,000	1.613	67,800	.794
83,900	217.9	74,500	2.709	76,400	1.075
92,000	690.3	85,300	4.490	89,400	1.668
94,400	991.3	94,700	7.190	103,500	2.759
96,700	1344.0	103,600	11.02	111,700	3.560
DC-200 AA-2036					
0	.46	0	.156	0	.092
14,000	2.55	19,900	1.16	10,500	.319
24,400	6.12	34,800	2.57	23,800	.698
35,000	14.2	49,800	5.26	36,300	1.314
45,600	34.4	62,900	10.1	51,300	2.491
56,000	93.6	77,500	20.7	55,800	3.08
66,600	286.5	91,900	43.3	70,500	5.10
70,400	Solid	104,200	89.5	83,200	7.93
		118,200	212.9	97,100	13.2
				108,800	20.0
				117,000	27.9
DC-200 AA-1945					
0	1.60	0	.503	0	.332
7,200	4.88	14,000	2.74	10,900	1.42
14,100	9.25	28,300	6.58	22,400	2.65
21,200	17.2	41,700	12.9	35,600	4.79
28,000	30.8	56,800	26.5	49,300	8.43
35,000	51.7	71,000	55.7	61,700	13.5
42,000	95.1	84,600	114.3	75,000	22.3
49,200	176.8	98,200	261.4	87,100	35.4
56,000	351.6	112,000	642.0	99,400	55.9
63,900	736.3			110,500	84.2
75,000	Solid			119,500	121.1

UCON 50-HB Series

25°C		98.9°C		150°C	
psi	Poises	psi	Poises	psi	Poises
UCON 50-HB-55					
0	.12	0	.021	0	.012
35,700	1.64	10,200	.042	7200	.018
42,400	2.29	20,200	.069	17,800	.027
52,700	4.11	32,200	.109	20,700	.033
63,200	7.15	42,700	.162	32,000	.046
73,500	12.1	53,500	.239	43,600	.065
84,000	19.8	63,300	.317	54,300	.088
94,200	38.5	74,900	.436	67,400	.126
104,000	90.5	85,000	.605	80,500	.170
		94,700	.837	88,500	.205
		106,200	1.14	97,400	.250
		115,600	1.458	109,200	.322
				115,500	.363
UCON 50-HB-170					
0	.59	0	.078	0	.037
7,200	1.44	9,500	.149	14,200	.082
21,500	4.28	21,300	.287	28,600	.150
34,900	12.7	32,200	.471	40,500	.229
49,200	37.3	42,700	.722	57,800	.393
59,700	79.3	54,300	1.200	71,700	.599
70,500	167.7	67,100	1.965	85,200	.872
80,600	337.3	79,200	3.058	97,500	1.20
90,700	636.1	91,400	4.770	109,000	1.61
		104,100	7.340		
UCON 50-HB-660					
0	2.57	0	.25	0	.103
11,200	7.86	21,500	1.12	14,600	.246
21,800	20.5	31,300	1.73	27,200	.426
31,500	46.5	43,700	2.83	42,500	.729
43,100	117.5	53,500	4.25	56,300	1.17
52,700	224.5	64,300	6.33	70,800	1.82
63,300	482.4	75,600	9.95	84,600	2.76
72,700	1076.0	84,800	14.1	96,700	3.82
		95,000	20.5	111,700	5.63
		106,700	31.0		
		112,700	38.0		

(Continued)

Table 1 (Contd-2). VISCOSITY-PRESSURE-TEMPERATURE RELATION

UCON LB Series						Di-Ester and Silicate Ester					
25°C		98.9°C		150°C		25°C		98.9°C		150°C	
psi	Poises	psi	Poises	psi	Poises	psi	Poises	psi	Poises	psi	Poises
UCON LB-70x						Plexol 201					
0	.19	0	.028	0	.0154	0	.14	0	.033	0	.0163
17,500	1.33	14,300	.076	10,400	.029	28,000	2.31	10,500	.068	8,200	.028
28,100	3.70	25,000	.137	22,000	.048	42,000	6.68	21,500	.132	19,300	.050
38,900	9.56	34,700	.225	31,500	.073	56,000	17.0	36,400	.261	30,000	.077
47,500	20.0	46,300	.366	44,300	.119	63,000	25.2	49,500	.470	42,000	.119
59,700	56.4	56,800	.568	60,200	.193	70,000	39.2	62,500	.792	55,100	.184
69,400	124.5	67,700	.930	73,300	.306	77,200	58.3	74,900	1.269	69,300	.286
81,300	309.0	78,400	1.453	89,000	.513	84,000	85.7	90,500	2.234	84,000	.446
90,500	635.7	91,300	2.537	105,500	.835	91,000	130.4	104,700	3.736	96,400	.624
		105,400	4.490	118,000	1.20	97,600	188.7			110,400	.890
		118,000	7.33			104,000	281.8			112,600	.941
UCON LB-170x						Orsil BF-1-S					
0	.56	0	.071	0	.034	0	.071	0	.020	0	.0128
7,200	1.93	14,800	.233	10,500	.071	35,200	2.26	10,500	.042	7,300	.021
17,500	6.36	28,000	.529	21,900	.132	46,000	4.69	21,400	.082	21,500	.042
28,100	22.9	43,000	1.278	32,200	.211	56,800	8.89	33,000	.151	33,500	.069
38,600	73.7	50,200	1.950	42,300	.319	66,900	17.0	43,000	.236	47,000	.113
49,200	237.2	56,400	2.70	55,400	.538	77,200	31.1	54,000	.383	63,000	.188
59,200	657.1	71,200	5.975	69,400	.914	87,800	55.6	64,400	.590	78,300	.306
65,500	1299.0	84,800	12.5	85,000	1.62	97,900	94.7	74,800	.913	91,600	.449
		98,200	22.8	97,500	2.55	104,600	132.6	84,600	1.316	104,000	.624
		105,400	32.9	110,700	3.95			94,200	1.842	111,000	.741
		112,300	46.8					106,700	2.833		
		118,000	61.1					112,100	3.268		
								115,000	3.600		
UCON LB-550x											
0	2.65	0	.205	0	.083						
7,900	7.32	17,500	.96	13,900	.224						
14,500	17.7	27,800	1.75	28,100	.464						
21,200	45.8	38,500	3.22	42,700	1.03						
28,000	111.5	50,200	6.46	56,000	1.71						
35,000	259.5	59,700	10.8	71,800	3.13						
42,000	601.7	71,400	21.5	85,200	5.20						
49,100	1370.0	81,300	37.8	98,200	8.48						
		94,800	85.7	108,000	12.1						
		105,500	153.2								
		111,500	211.5								
		115,500	253.5								

Table 2. CONSTANTS A, B, AND C PERTAINING TO AN EMPIRICAL EQUATION

$$\log \eta = A + \frac{B(P + C)}{T^3}$$

Lubricants	Constants		
	A	B	C
1010 grade mineral oil	-3.0958	23.5420	1.0486
SAE 10 w mineral oil	-2.9117	22.1327	2.1293
SAE 20 w mineral oil	-2.9232	24.4055	2.2960
SAE 30 w mineral oil	-3.1087	31.3807	2.2326
SAE 60 w mineral oil	-2.5293	24.0568	2.9055
UCON 50-HB-55	-2.9432	10.8981	2.3752
UCON 50-HB-170	-2.6079	13.0419	2.9015
UCON 50-HB-660	-2.2368	13.1555	3.5622
UCON LB-70x	-3.3450	14.5687	2.5832
UCON LB-170x	-2.8319	17.2088	2.9273
UCON LB-550x	-2.5324	18.8540	2.9951
Silicone DC-200 AA 2136	-2.0826	14.5644	-0.6357
Silicone DC-200 AA 2036	-1.4053	17.5451	-0.1585
Silicone DC-200 AA 1945	-0.8282	17.9018	-0.2089
Plexol 201	-2.7084	13.0745	1.8146
Orsil BF-1-S	-2.6969	13.0697	1.1813

Estimation of Viscosities of Oil Films Under Contact Conditions and Their Relation to Load Carrying Capacity

The data on viscosity - pressure - temperature characteristics of oils permitted a further study of the relation between viscosity and load carrying capacity. Several different approaches were used in these studies.

First and probably the most direct approach used consisted of estimating viscosity of the oil films between the meshing gear teeth and relating it to the corresponding load carrying capacity. However, the exact pressure and temperature within the contacting film are unknown and there are no methods for their determinations. The viscosity calculations were based, therefore, on an assumption that the pressure within the contacting film was equivalent, or proportional, to the Hertz Stress between the meshing gear teeth. The equation used was:

$$P_a = \sqrt{\frac{0.35w\ (1/d_1 + 1/d_2)}{L\ (1/E_1 + 1/E_2)}}$$

Where: P_a = Hertz Stress at addendum of pinion - psi
W = Load at scoring - pounds
d_1 = radius of curvature at addendum of pinion - in
d_2 = radius of curvature at dedendum of gear in
L = width of gear face - in
$E_1 = E_2$ = modulus of elasticity.

The viscosities of the oils under such pressures and temperatures of 25 °C, 100°C, 150°C, 200°C and 250°C were calculated using the empirical equation:

$$\log \eta = A + B\,\frac{(P + C)}{T^3}$$

Where P = pressure calculated by the Hertz formula
T = absolute temperature
A, B and C = constants (see Table 2).

The results of these calculations are presented in Tables 3, 4 and 5 and graphically in Figure 14.

Discussion of Results. The definition of gear failure by scoring (page 3, Summary Report S-13605) requires the complete destruction of the lubricating film and the establishment of metal to metal contact between the rubbing surfaces. It may appear, therefore, that an association of viscosity with score load is without meaning. However, since the experimental data have shown that the phenomenon of scoring is an instantaneous one for all unreactive oils and as it is the condition immediately prior to scoring that is of interest, the attempt to derive conclusions from the association is justifiable.

As was mentioned above, the exact pressure within the contacting film is unknown and the evaluation of the viscosity was made on an assumption that the maximum pressure within the film is equal or proportional to the Hertz pressure. The Hertz equation, however, was developed for the direct metal to metal contact under static conditions. In gears the teeth mesh with rolling and sliding motion which may considerably alter the pressure. Furthermore, the supporting effect of the incoming part of the oil film should decrease the pressure as evaluated by the Hertz equation. Evidence for this is seen in the fact that the viscosity of the film, as evaluated using the full Hertz pressure, is astronomically high. For example, at a score load of 50 pounds, which corresponds to 366,000 psi Hertz pressure, the viscosity of SAE 30 mineral oil at 100°C is equal to 4.11×10^{14} poises, which value for all practical purposes represents a "solid". On account of this, the viscosities at one-fourth of the Hertz Stress were also calculated. The fraction of one-fourth was chosen because it gives the pressures less than 100,000 psi. In the graphical form only the data for one-fourth of the Hertz Stress and temperatures of 100, 150, 200 and 250°C are presented.

The graphs given in Figure 14 show for four temperature levels the relation existing between viscosity of the lubricants and pressure (expressed in terms of score load). Three families of lubricants are presented: mineral oils, silicones and polyglycols. It is of interest that each family of lubricants is represented by a group of points forming well-defined bands. The inclusion of the data for all speeds used explains why the relation between viscosity and load carrying capacity is shown by bands, not lines.

These plots show that the relative positions of the families of oils depend upon the temperature. For example, at an assumed film temperature of 100°C the bands for the silicone and mineral oils are similar, and the band for the polyglycol oils lies distinctly to the right, or at positions of higher loads for equal viscosities. For temperatures above 100°C the bands for mineral oils move to the right of the band for the silicone oils, and finally, if the temperature of the oil films is 250°C, or over, mineral oil viscosities for the same pressure are even less than the polyglycols.

No attempts were made to establish a mathematical relation between viscosity of the films and load carrying capacity, because these studies are based on assumed values of pressure and temperature. However, these studies have demonstrated the temperature dependence of viscosity - score load relations.

Conclusions. The conclusions deduced from the studies are: (1) lubricants of a similar chemical nature form separate bands or groups on plots of viscosity of the films as a function of score loads, (2) the position of these bands relative to one another is temperature dependent, (3) a knowledge of the actual oil film temperatures will be necessary for a real understanding of the relationship between viscosity and load carrying capacity.

The Effect of Viscosity - Pressure - Temperature Characteristics of Oils on Load Carrying Capacity. Analytical Approach

The foregoing section presents a direct approach to examining the relations between the measured viscosity - temperature - pressure characteristics of lubricants and their load carrying capacities. An alternative attack involving an analytical re-examination of the load carrying - speed - viscosity relations is as follows:

Theoretical Considerations. On first consideration a quantitative description of the load carrying capacity of lubricants in spur gear systems in relatively fundamental terms appears inordinately difficult. Direct measurements of the times, temperatures and pressures involved between surfaces of meshing gear teeth have not been successful. Calculations of these quantities to date have involved too many assumptions to have much worth, as was well demonstrated by the preceding studies. However, a closer examination of the situation of two surfaces approaching one another as with meshing gear teeth suggests a useful parameter for looking at such systems.

The thickness of the lubricant film normally separating gear teeth surfaces should be proportional to the viscosity η of the film under contact conditions, and inversely proportional to the "contact" time t and pressure p. Scoring with uncompounded lubricants must occur when the lubricant film between the gear teeth reaches a limiting thickness so that a significant proportion of the metal asperities are contacting. Hence, at, or just prior to scoring conditions, the film thickness, or the average distance between the gear surfaces, is constant. Therefore, if the load carrying capacity of uncompounded lubricants is essentially a "hydrodynamic" phenomenon, the dimensionless parameter $\frac{\eta}{p \times t}$ should be constant at scoring.

Considering $\frac{\eta}{p \times t}$ to be constant at scoring conditions, then $p \times t = K.\eta$ should be an excellent parameter for correlating gear load carrying data. Furthermore, it is not necessary to calculate hypothetical pressures and contact times for a particular gear system as $p \times t$ is directly proportional to the applied load, divided by the velocity, i.e., $p \times t = K_2 \frac{BL}{S}$ where BL = beam load and S = rpm. From this it follows that viscosity divided by $\frac{BL}{S}$ should also be a constant at scoring conditions. In the following discussion spur gear load carrying data are examined in terms of $\frac{BL}{S}$ which has the dimensions of viscosity when beam load is converted to pressure.

In Figures 15 to 19, the load carrying capacity data (see Tables 8 and 9 of Summary Report S-13605 and Tables 8, 9 and 10 of Summary Report S-13649) for various mineral and synthetic oils are presented in terms of the relations between the viscosity function $\frac{BL}{S}$ and S the speed of operation in rpm. From examination of these curves several observations of interest can be made.

1. In the 1000 - 5000 rpm region, the behavior of the lubricants examined can be generally described in terms of expressions of the form

$$\frac{BL}{S} = \alpha S^n$$

2. Within this 1000 - 5000 rpm region and for a given class of lubricants such as the mineral oils, there is a quite regular change in the slopes of the $\frac{BL}{S}$ curves with increasing viscosity or molecular weight. This is demonstrated in Figure 20 for mineral oils where log α and η calculated from the curves of Figure 15 have been plotted against each other. The indicated relationship, $\log \alpha = -(2.65n + 0.6)$, means that the mineral oil

load carrying data generally can be represented by exponential curves originating from a common point.[a] It would appear that a similar situation holds for the various synthetic oils although there are insufficient data to firmly establish this.

A useful outcome of this finding for mineral oils is that it allows the prediction of the load carrying capacity curve with speed from a single determination. For example

$$\frac{BL}{S} = \alpha S^n$$

$$\log BL = \log \alpha + (n + 1) \log S$$

$$= -(2.65\eta + 0.6) + (\eta + 1) \log S$$

From the measurement of score load at any speed in the 1000 - 5000 rpm region, η can be determined and score loads at other speeds predicted.

3. At speeds above 5000 rpm there is a region of markedly different behavior where the viscosity function BL/S actually increases with speed. In the past this departure from what might be called hydrodynamic behavior has been attributed to the onset of visco-elastic phenomena. The departures become less with the more viscous higher molecular weight lubricants. The sharp delineation of this region of discontinuity should make it possible to more clearly define the phenomenon. At least, quantitative distinctions between lubricants can be made.

The log α's and η's for the 1000 - 5000 rpm region calculated from the curves of Figures 15 to 19 are presented in Table 5. Here, the values for SAE 20 mileral oil, UCON 50-HB-400 and UCON LB-170x were obtained from the indicated relationships between log α and η. In the case of the silicones, there is considerable scatter in the data and log α and η have been calculated only for the 10 centistokes fluid for which there is a three point curve.

Relation of Load Carrying Capacity to Lubricant Viscosity Characteristics. As demonstrated above, use of a viscosity function concisely and clearly illustrates the similarities end differences between uncompounded lubricants. Moreover, it has permitted describing the load carrying capacity of such lubricants over a considerable range of speeds in terms of just two components: α which can be considered generally to determine the load carrying level and η, the effect of speed on load carrying capacity. Further, for mineral oils at least, α can be described in terms of η with considerable accuracy. However, so far, this is still only indirect evidence of the importance of viscosity. More conclusive would be the finding of simple relations between the expressions describing load carrying capacity and those describing viscosity as a function of temperature and pressure.

The viscosities of most of the lubricants considered in the foregoing section have been determined over a broad range of temperatures (25°C to 150°C) and pressures up to 120,000 psi. The results, as was shown in the preceding

a) The origin in the case of mineral oils corresponds to 100 lb bear load at 450 rpm.

section, can be described quite well by empirical expression of the form

$$\log \eta = A + B \frac{(P + C)}{T^3}$$

where A, B and C are constants for any one lubricant and P and T are pressure and temperature respectively. However, we have assumed $\frac{BL}{S} = K\eta$ and hence $\log K\eta = \log \alpha + n \log S$. The similarity of the two expressions is obvious. If the bulk viscosity characteristics of lubricants were all important in determining their load carrying capacity one would expect to find a relation between first A's and log α's and second, B's and n's. The constants C are of the least importance in the viscosity relationship.

Refering to the viscosity characteristics of lubricants, the determined A, B and C constants for mineral oils, silicones and UCON 50-HG and LB have been plotted in Figures 21, 22, 23 and 24 against lubricant viscosity (cs at 100°F). The family similarities are obvious with the exception that the SAE 30 mineral oil differs significantly from the other mineral oils. This difference shows up in the final comparisons.

Considering the possible relation between log α and A, the available data are plotted in Figure 25. It would appear that the expected relationship is found within a given class of lubricants but that the differences between classes still are not resolved. A possible exception is the two classes of UCONs. A similar situation is apparent when n is plotted against B as in Figure 26. Here, however, there is less similarity between the UCON fluids.

Conclusions. The hydrodynamic concept of the operation of loaded spur gears has provided a parameter which more clearly demonstrates the similarities and differences between uncompounded lubricants. It makes it possible to describe the load carrying capacity of a lubricant over a considerable range of speeds in terms of just two constants. It permits the quantitative delineation of the region of discontinuity in the viscosity - speed relation. These advances are additional evidence of the importance of the viscosity characteristics of lubricants in explaining load carrying capacity.

Attempts to relate load carrying capacity directly to lubricant bulk viscosity characteristics were only partially successful. These studies indicate that in order to account for the differences in performance of various classes of lubricants, liquid properties in addition to just viscosity - temperature - pressure characteristics will have to be considered.

 Tables 3, 4 and 5 follow

Table 3. VISCOSITIES OF MINERAL OILS AT SCORE LOADS

Oils	Bulk Viscosity-cs at 100°F	Bulk Viscosity-cs at 210°F	Speed rpm	At Scoring: Beam Load lb	At Scoring: Hertz Stress psi	Viscosity at Full Hertz Stress - Poises: 25°C	98.9°C	150°C	200°C	250°C	1/4 of Full Hertz Stress psi	Viscosity at 1/4 Hertz Stress - Poises: 25°C	98.9°C	150°C	200°C	250°C
1010 Mineral Oil	9.97	2.50	1000	36	301×10^3	1.78×10^{17}	7.27×10^7	4.70×10^4	556.5	30.62	75,250	2750.0	5.611	.694	.195	.0849
			3000	12	179×10^3	6.14×10^9	1.00×10^4	101.42	7.55	1.271	44,750	37.42	.6202	.154	.066	.0038
			5000	4	103×10^3	1.03×10^5	42.34	2.72	.517	.175	25,750	2.57	.1564	.051	.034	.0029
			10000	4	103×10^3	1.03×10^5	42.34	2.72	.517	.175	25,750	2.57	.1564	.051	.034	.0029
			15000	8	145×10^3	5.09×10^6	889.85	21.54	2.27	.524	38,250	14.27	.394	.1016	.053	.0033
			20000	10	162×10^3	5.58×10^8	3052.0	49.81	4.14	.816	40,500	20.57	.456	.1024	.057	.0034
SAE 10W Mineral Oil	34.60	5.41	1000	50	366×10^3	8.28×10^{20}	6.89×10^9	1.19×10^8	6326.0	206.4	91,500	1.345×10^5	50.15	3.559	.708	.2465
			3000	10	162×10^3	1.53×10^9	6295.0	93.33	7.32	1.388	40,500	156.87	1.593	.336	.131	.00706
			5000	8	145×10^3	1.61×10^8	1976.0	42.45	4.18	.917	38,250	116.56	1.366	.302	.121	.00665
			10000	10	162×10^3	1.53×10^9	6295.0	93.33	7.32	1.388	40,500	156.87	1.593	.336	.131	.00706
			15000	16	205×10^3	4.54×10^{10}	1.18×10^5	684.3	30.46	3.985	51,250	651.3	3.313	.551	.185	.00919
			20000	24	251×10^3	2.01×10^{14}	2.71×10^8	5771.4	139.9	12.310	62,750	2986.0	7.255	.939	.279	.01218
SAE 20 Mineral Oil	74.81	8.70	3000	12	179×10^3	3.49×10^{11}	1.01×10^5	613.8	27.97	3.716	44,750	1037.5	4.157	.637	.205	.00979
			10000	16	205×10^3	1.57×10^{13}	7.23×10^6	5841.1	72.53	7.518	51,250	2761.0	6.85	.898	.262	.01175
			15000	24	251×10^3	1.29×10^{16}	2.28×10^7	2.43×10^4	388.5	25.97	62,750	14,850.0	13.50	1.616	.399	.01605
			20000	24	251×10^3	1.29×10^{16}	2.28×10^7	2.43×10^4	388.5	25.97	62,750	14,850.0	13.50	1.616	.399	.01605
SAE 30 Mineral Oil	132.2	10.60	1000	50	366×10^3	2.41×10^{30}	4.11×10^{14}	1.81×10^9	1.05×10^6	8063.0	91,500	9.9×10^7	1256.0	26.68	2.63	.5781
			3000	24	251×10^3	1.01×10^{21}	6.12×10^9	9.48×10^5	4730.0	217.6	62,750	4.72×10^5	78.88	4.11	.690	.0215
			5000	16	205×10^3	1.79×10^{17}	7.18×10^7	4.62×10^4	544.6	29.91	51,250	50,370.0	25.15	1.89	.396	.0142
			10000	20	219×10^3	2.48×10^{18}	2.78×10^8	1.05×10^5	1005.1	48.66	54,750	99,977.0	35.43	2.38	.467	.0161
			15000	24	251×10^3	1.01×10^{21}	6.12×10^9	9.48×10^5	4730.0	217.6	62,750	4.72×10^5	78.88	4.11	.690	.0215
			20000	24	251×10^3	1.01×10^{21}	6.12×10^9	9.48×10^5	4730.0	217.6	62,750	4.72×10^5	78.88	4.11	.690	.0215
SAE 60 Mineral Oil	393.0	26.0	1000	70	429×10^3	8.48×10^{27}	4.28×10^{18}	5.97×10^8	6.95×10^5	10,150.0	107,250	6.32×10^7	1864.2	54.12	6.37	1.601
			3000	44	338×10^3	1.73×10^{22}	5.04×10^{10}	6.10×10^6	2.62×10^4	1052.0	84,500	2.45×10^6	352.1	17.43	2.84	.0893
			5000	36	301×10^3	7.69×10^{19}	3.26×10^9	9.47×10^5	6911.0	382.3	75,250	6.48×10^5	177.4	11.72	2.03	.0676
			10000	40	317×10^3	8.43×10^{20}	1.01×10^{10}	2.12×10^6	1.02×10^4	585.6	79,250	1.20×10^6	243.1	13.55	2.36	.0755
			15000	44	338×10^3	1.73×10^{22}	5.04×10^{10}	6.10×10^6	2.62×10^4	1052.0	84,500	2.45×10^6	352.1	17.43	2.84	.0893
			20000	40	317×10^3	8.43×10^{20}	1.01×10^{10}	2.12×10^6	1.02×10^4	585.6	79,250	1.20×10^6	243.1	13.55	2.36	.0755

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Table 4. VISCOSITIES OF UCON OILS AT SCORE LOAD

	Bulk Viscosity cs		Speed rpm	At Scoring		Viscosity at Full Hertz Stress - Poises					1/4 of Full Hertz Stress psi	Viscosity at 1/4 Hertz Stress - Poises				
	100°F	210°F		Beam Load lb	Hertz Stress psi	25°C	98.9°C	150°C	200°C	250°C		25°C	98.9°C	150°C	200°C	250°C
UCON LB - 70X	12.14	2.82	3000	12	179×10^{3}	7.11×10^{5}	74.85	3.327	.508	.148	44,750	5.87	.185	.055	.0266	.0170
			5000	8	145×10^{3}	3.67×10^{4}	16.25	1.017	.242	.086	38,250	3.33	.135	.045	.0236	.0153
			10000	20	219×10^{3}	2.33×10^{7}	499.8	11.01	1.0198	.284	54,750	11.40	.283	.075	.034	.0201
			20000	16	205×10^{3}	6.86×10^{6}	240.0	7.361	.896	.226	51,250	11.03	.242	.067	.031	.0189
UCON LB - 170X	40.28	7.39	3000	28	271×10^{3}	1.51×10^{12}	2.42×10^{5}	1016.9	46.36	6.813	67,750	1254.0	5.05	.79	.258	.123
			5000	32	290×10^{3}	1.01×10^{13}	6.62×10^{5}	2398.8	77.45	8.531	72,500	2030.0	6.58	.94	.289	.134
			10000	28	271×10^{3}	1.51×10^{12}	2.42×10^{5}	1016.9	46.36	6.813	67,750	1254.0	5.05	.79	.258	.123
			20000	28	271×10^{3}	1.51×10^{12}	2.42×10^{5}	1016.9	46.36	6.813	67,750	1254.0	5.05	.79	.258	.123
UCON LB - 550X	125.5	19.58	3000	70	429×10^{3}	4.15×10^{21}	2.40×10^{10}	3.685×10^{6}	1.82×10^{4}	56.69	107,250	6.92×10^{5}	128.9	11.14	2.05	.68
			5000	-	-	-	-	-	-	-	-	-	-	-	-	-
			10000	48	353×10^{3}	7.84×10^{17}	2.92×10^{8}	1.835×10^{5}	2132.0	10.89	88,250	8.35×10^{4}	61.48	5.32	1.52	.39
			20000	44	338×10^{3}	1.44×10^{17}	1.02×10^{8}	1.001×10^{5}	1396.0	9.05	84,500	5.47×10^{4}	49.47	4.58	1.09	.44
UCON 50-HB - 55	8.36	2.27	3000	10	162×10^{3}	4.18×10^{3}	8.331	1.001	.281	.122	40,500	1.517	.141	.063	.039	.028
			5000	6	127×10^{3}	4.25×10^{2}	2.57	.453	.159	.0803	31,750	1.168	.105	.049	.032	.024
			10000	12	179×10^{3}	1.26×10^{4}	14.74	1.485	.371	.150	44,750	1.998	.163	.069	.042	.030
			20000	20	219×10^{3}	1.75×10^{5}	56.44	3.700	.713	.249	54,750	3.835	.227	.087	.049	.039
UCON 50-HB - 170	35.31	7.56	3000	23	217×10^{3}	1.00×10^{9}	7.14×10^{3}	127.3	10.111	.227	67,750	130.57	2.087	.49	.21	.121
			5000	24	251×10^{3}	2.12×10^{8}	3.20×10^{3}	73.72	7.55	.170	62,750	98.97	1.76	.45	.20	.115
			10000	32	290×10^{3}	4.45×10^{9}	1.55×10^{4}	213.8	16.176	.299	72,500	184.8	2.44	.55	.23	.129
			20000	36	301×10^{3}	1.01×10^{11}	2.38×10^{4}	288.7	20.05	.351	75,250	234.1	2.75	.61	.24	.134

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Table 5. VISCOSITIES OF SILICONES AND DI-ESTER AT SCORE LOAD

Oils	Bulk Viscosity cs		Speed	At Scoring		Viscosity at Full Hertz Stress - Poises					1/4 of Full Hertz Stress	Viscosity at 1/4 Hertz Stress - Poises				
	100°F	210°F	rpm	Beam Load lb	Hertz Stress psi	25°C	98.9°C	150°C	200°C	250°C	psi	25°C	98.9°C	150°C	200°C	250°C
Silicone DC-200 AA-2126	8.099	3.545	3000	3	145×10^3	5.69×10^4	83.44	9.113	2.39	.996	38,250	5.18	0.695	0.351	0.233	0.178
			5000	4	103×10^3	1.46×10^3	12.69	2.53	0.96	.506	25,750	1.74	.397	.240	.177	.145
			10000	8	145×10^3	5.69×10^4	83.44	9.113	2.39	.996	38,250	5.18	.695	.351	.233	.178
			20000	10	162×10^3	2.50×10^5	179.1	15.30	3.46	1.310	40,500	5.30	.769	.376	.245	.184
Silicone DC-200 AA-2036	36.77	15.30	3000	36	301×10^3	1.64×10^{13}	4.02×10^6	2.29×10^4	1006.4	130.7	75,250	833.7	20.73	5.73	2.67	1.62
			5000	28	271×10^3	5.58×10^{11}	7.95×10^5	7600.0	457.6	72.93	67,750	379.3	13.51	4.35	2.19	1.40
			10000	16	205×10^3	6.87×10^8	2.25×10^4	673.0	79.57	20.22	51,250	67.1	3.53	2.37	1.42	1.02
			20000	20	219×10^3	2.99×10^9	4.79×10^4	1011.9	101.56	26.55	54,750	231.0	10.46	3.66	1.94	1.28
Silicone DC-200 AA-1945	136.5	55.74	3000	50	366×10^3	1.012×10^{17}	7.30×10^8	1.02×10^6	2.[illegible]9 x 10^4	1987.0	91,500	25188.0	223.1	44.67	16.99	9.01
			5000	24	251×10^3	5.113×10^{11}	1.28×10^6	1.61×10^4	1013.4	203.1	62,750	892.3	39.99	13.93	7.34	4.86
			10000	20	219×10^3	1.658×10^{10}	2.20×10^5	4833.0	486.0	107.2	54,750	378.7	25.72	10.32	5.44	4.08
			20000	24	251×10^3	5.113×10^{11}	1.28×10^6	1.61×10^4	1013.4	203.1	62,750	892.3	39.99	13.93	7.34	4.86
Plexol 201	12.78	3.39	3000	30	284×10^3	6.86×10^8	5,233.0	95.61	8.51	.175	71,000	39.53	.986	.286	.132	.0801
			5000	16	205×10^3	1.42×10^6	217.32	10.09	1.81	.0558	51,250	8.47	.443	.160	.089	.0602
			10000	26	261×10^3	1.01×10^8	2,073.0	50.95	5.30	.125	65,250	25.30	.782	.240	.12	.0737
			20000	24	251×10^3	5.19×10^7	1,585.4	38.74	4.46	.1087	62,750	20.83	.707	.224	.112	.0711
Orsil BF-1-S	6.76	2.23	3000	36	301×10^3	1.28×10^9	7,320.0	101.94	10.02	.1851	75,250	27.63	.828	.252	.123	.0766
			5000	24	251×10^3	2.58×10^7	990.8	30.85	2.62	.0975	62,750	10.40	.501	.179	.096	.0639
			10000	32	290×10^3	5.44×10^8	4,704.0	89.66	8.19	.1716	72,500	21.78	.73	.230	.116	.0733
			20000	30	280×10^3	2.49×10^8	3,145.0	68.22	6.74	.1485	70,000	17.55	.67	.220	.110	.0710

Table 6. LOAD CARRYING CONSTANTS FOR LUBRICANTS

Lubricants		Equation	Constants	
			log α	n
Mineral Oils	1010 grade	$\log \alpha = -2.65\eta - 0.6$	5.82	-2.41
	SAE 10w		5.60	-2.32
	SAE 20		5.15	-2.17
	SAE 30		4.29	-1.84
	1065 grade		4.25	-1.83
	SAE 60		3.17	-1.44
Silicones	DC200 AA 2126		5.39	-2.29
UCON 50-HB	55	$\log \alpha = -2.83\eta - 1.17$	4.23	-1.91
	170		2.79	-1.37
	400		1.90	-1.08
UCON LB	70 x	$\log \alpha = -2.64\eta - 0.58$	5.08	-2.14
	170 x		3.69	-1.62
	550 x		2.78	-1.27
Di-Esters	Plexol 201		4.58	-1.91
	Plexol 244		5.59	-2.23

 Figures 14 through 26 follow

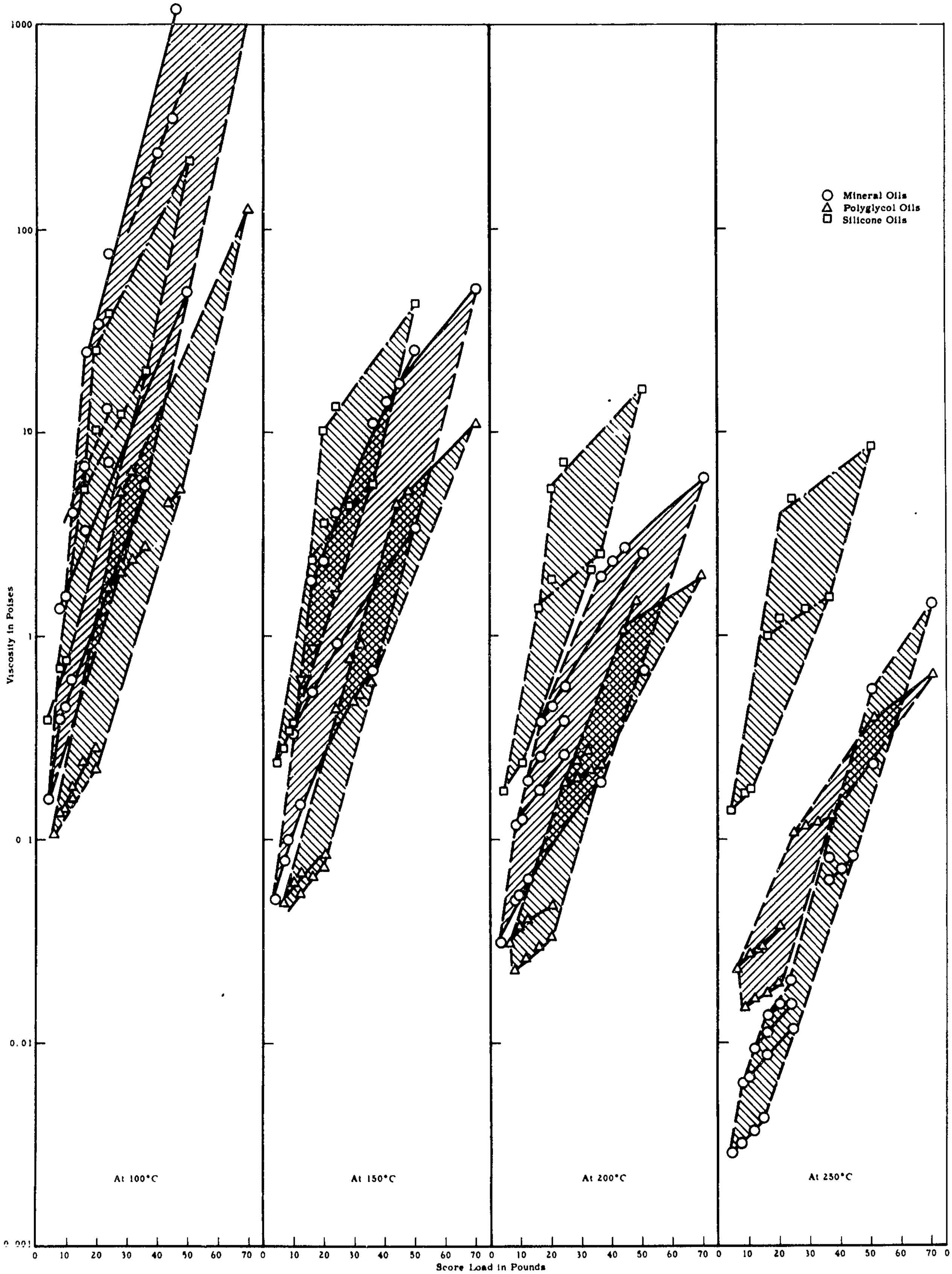

Figure 14. RELATION BETWEEN VISCOSITY OF CONTACTING FILMS AND LOAD CARRYING CAPACITY FOR MINERAL OILS, SILICONES AND POLYGLYCOLES

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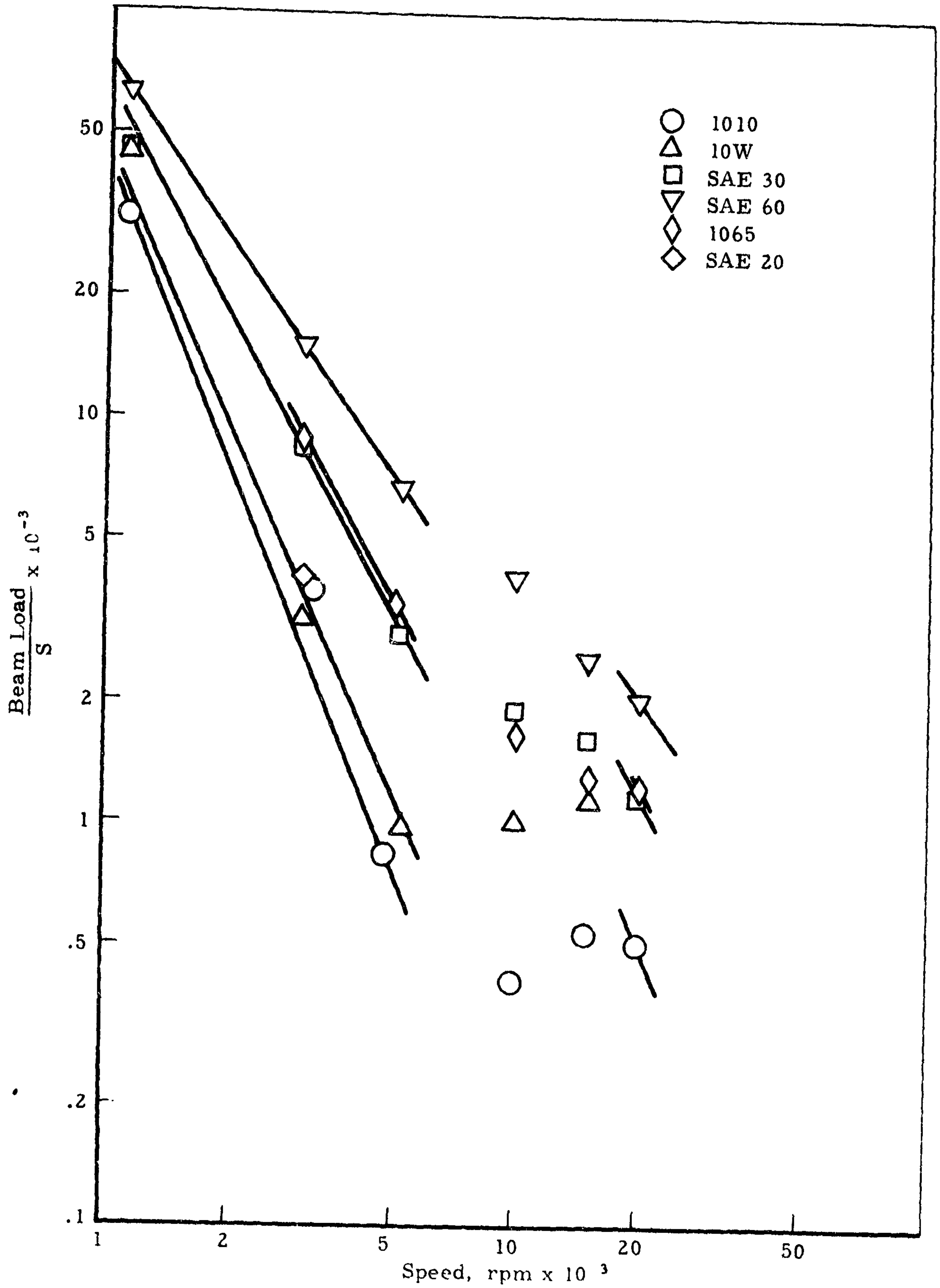

Figure 15. VISCOSITY FUNCTION BL/S VS SPEED FOR MINERAL OILS

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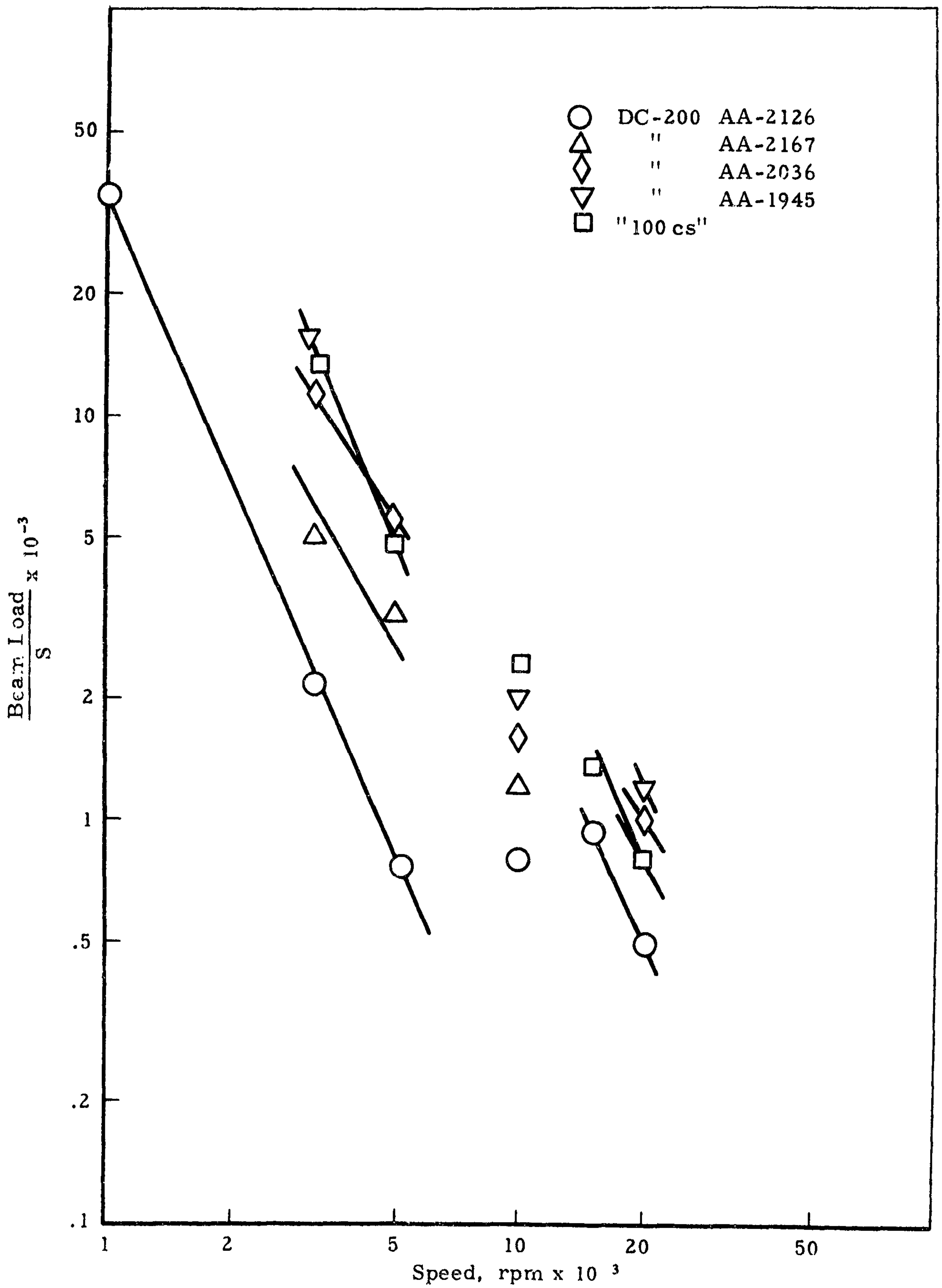

Figure 16. VISCOSITY FUNCTION BL/S VS SPEED FOR SILICONES

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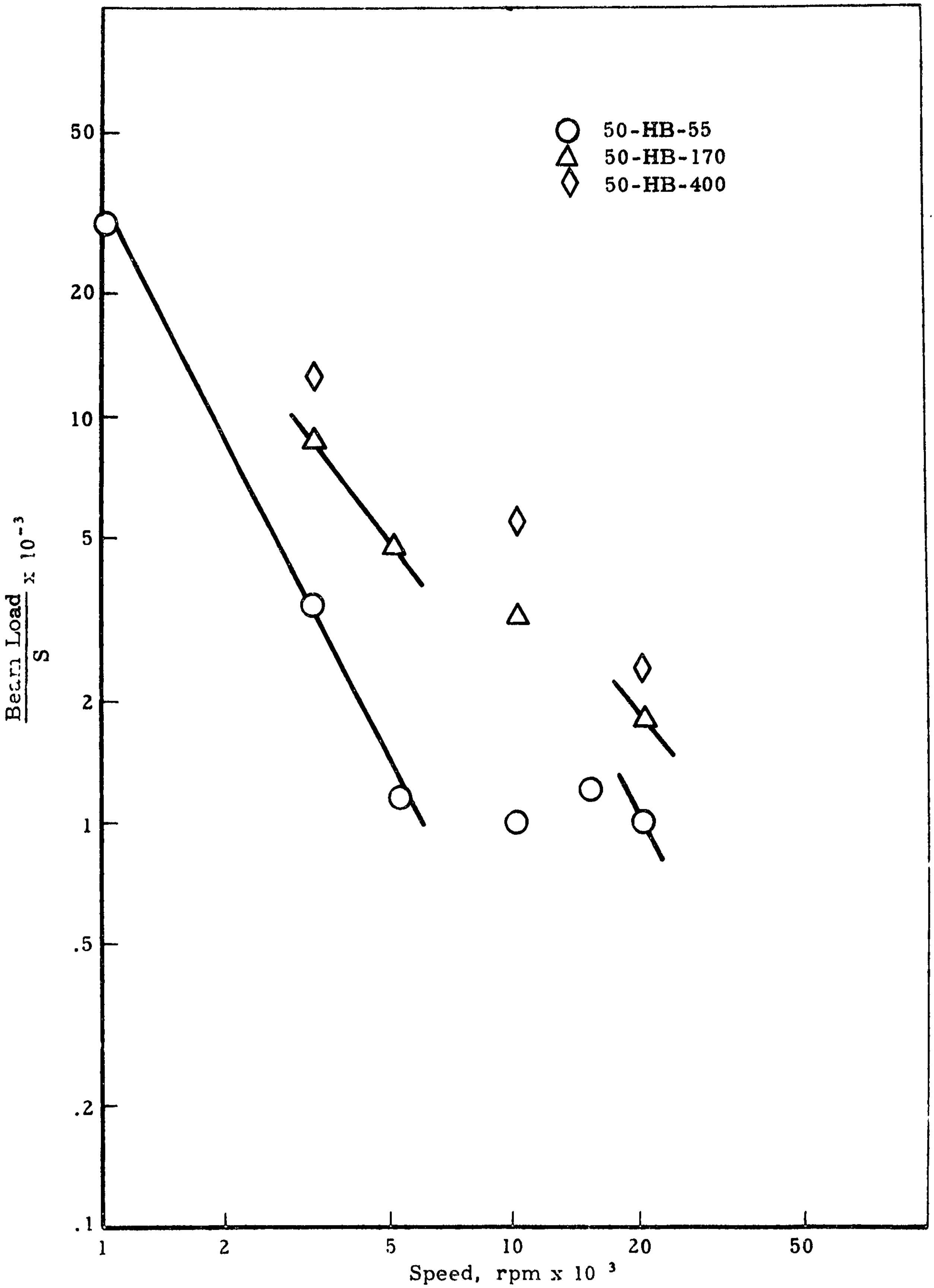

Figure 17. VISCOSITY FUNCTION BL/S VS SPEED FOR UCON 50-HB (SERIES)

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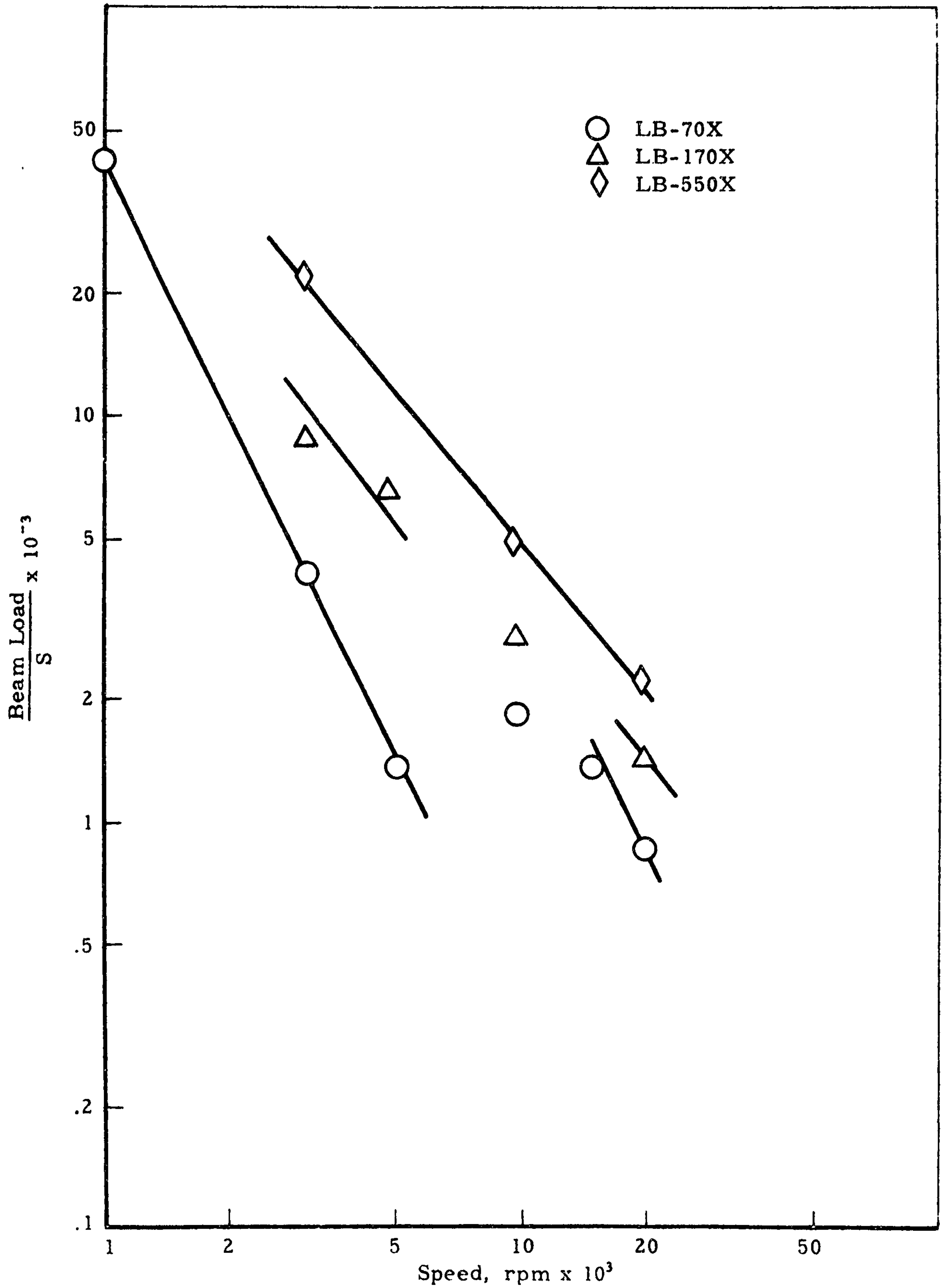

Figure 18. VISCOSITY FUNCTION BL/S VS SPEED FOR UCON LB (SERIES)

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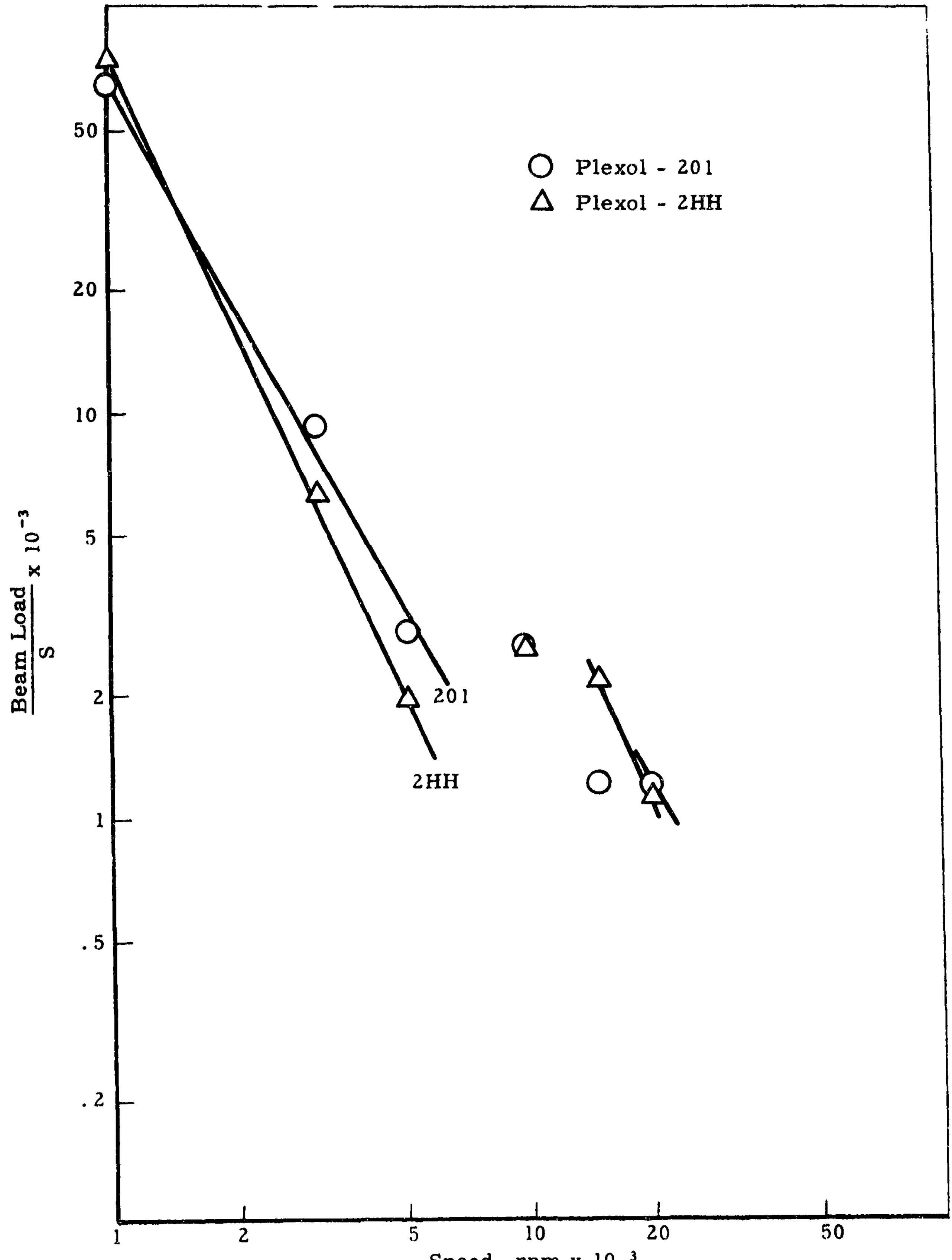

Figure 19. VISCOSITY FUNCTION BL/S VS SPEED FOR DI-ESTERS PLEXOLS

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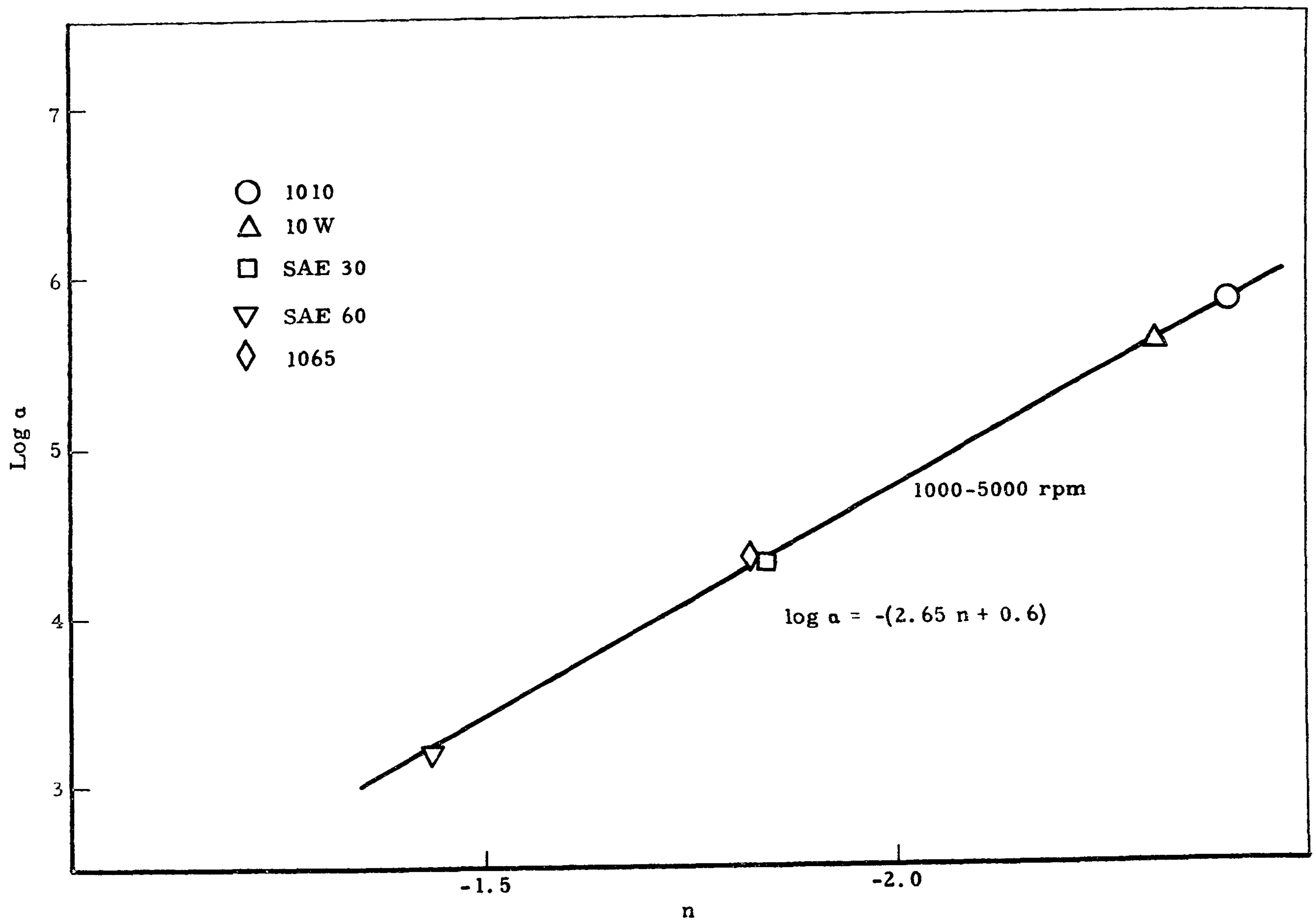

Figure 20. RELATION BETWEEN LOG a AND n OF THE EQUATION $BL/S = aS^n$ FOR MINERAL OILS

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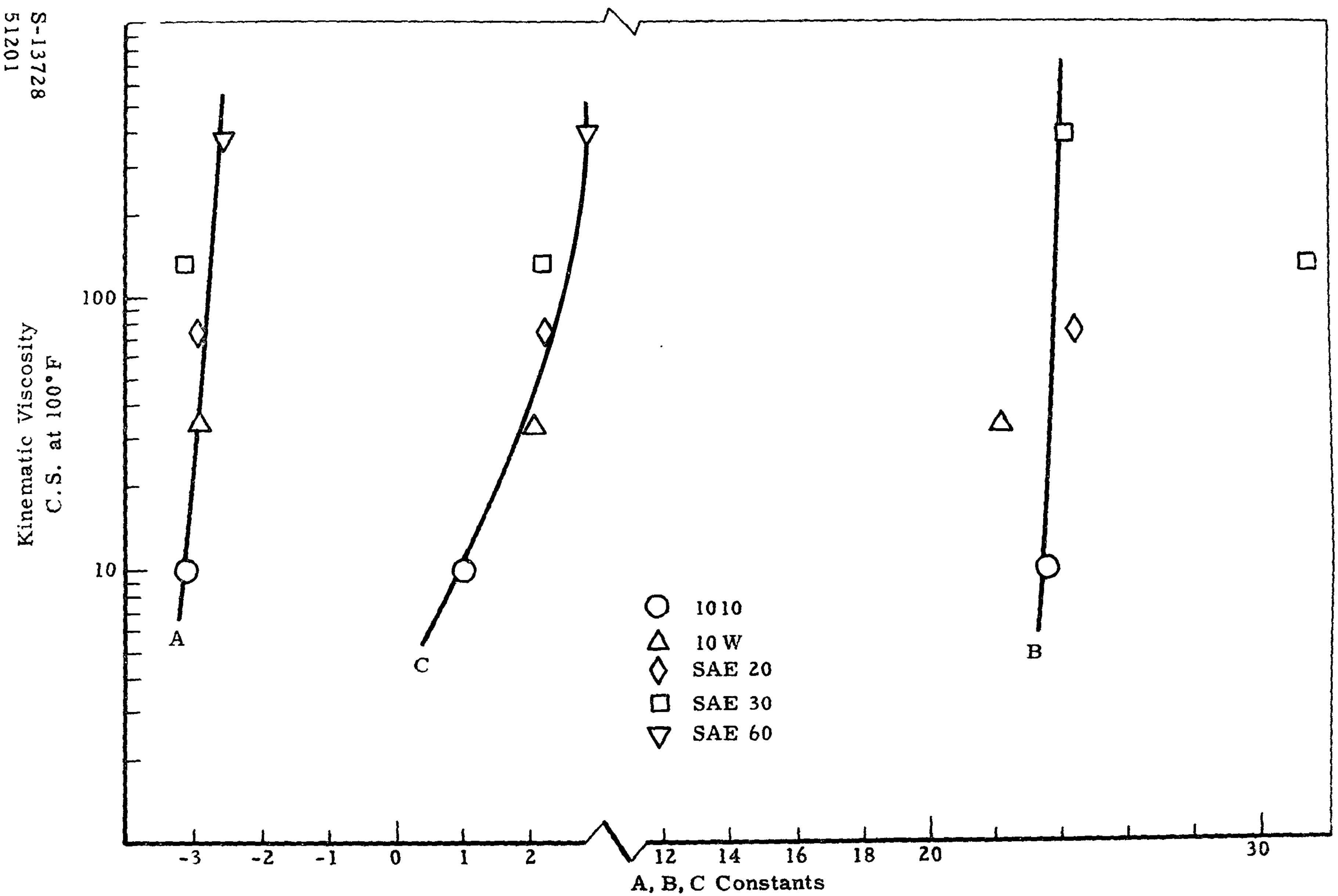

Figure 21. RELATION BETWEEN VISCOSITY AND CONSTANTS A, B AND C FOR MINERAL OILS

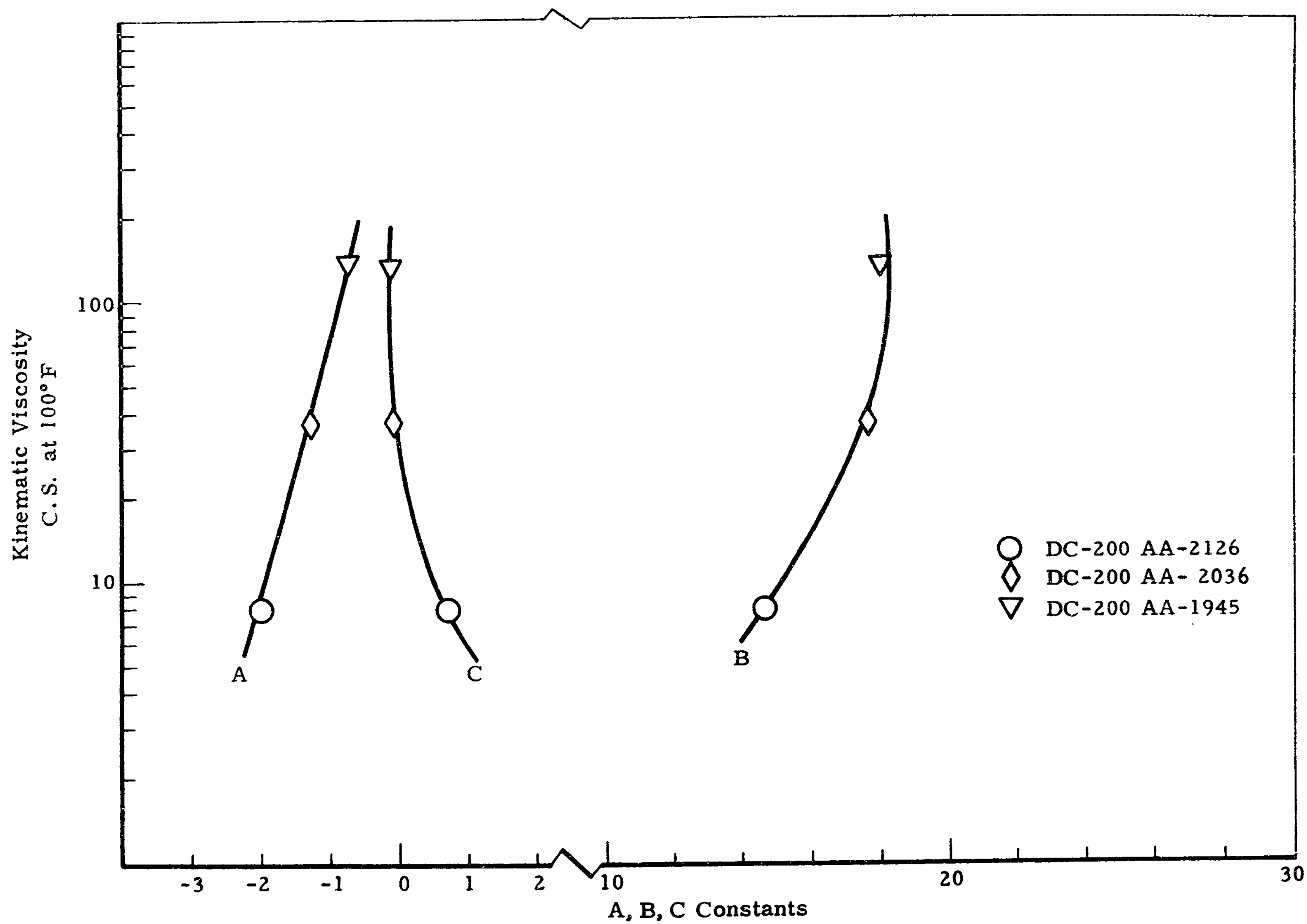

Figure 22. RELATION BETWEEN VISCOSITY AND CONSTANTS A, B AND C FOR SILICONES

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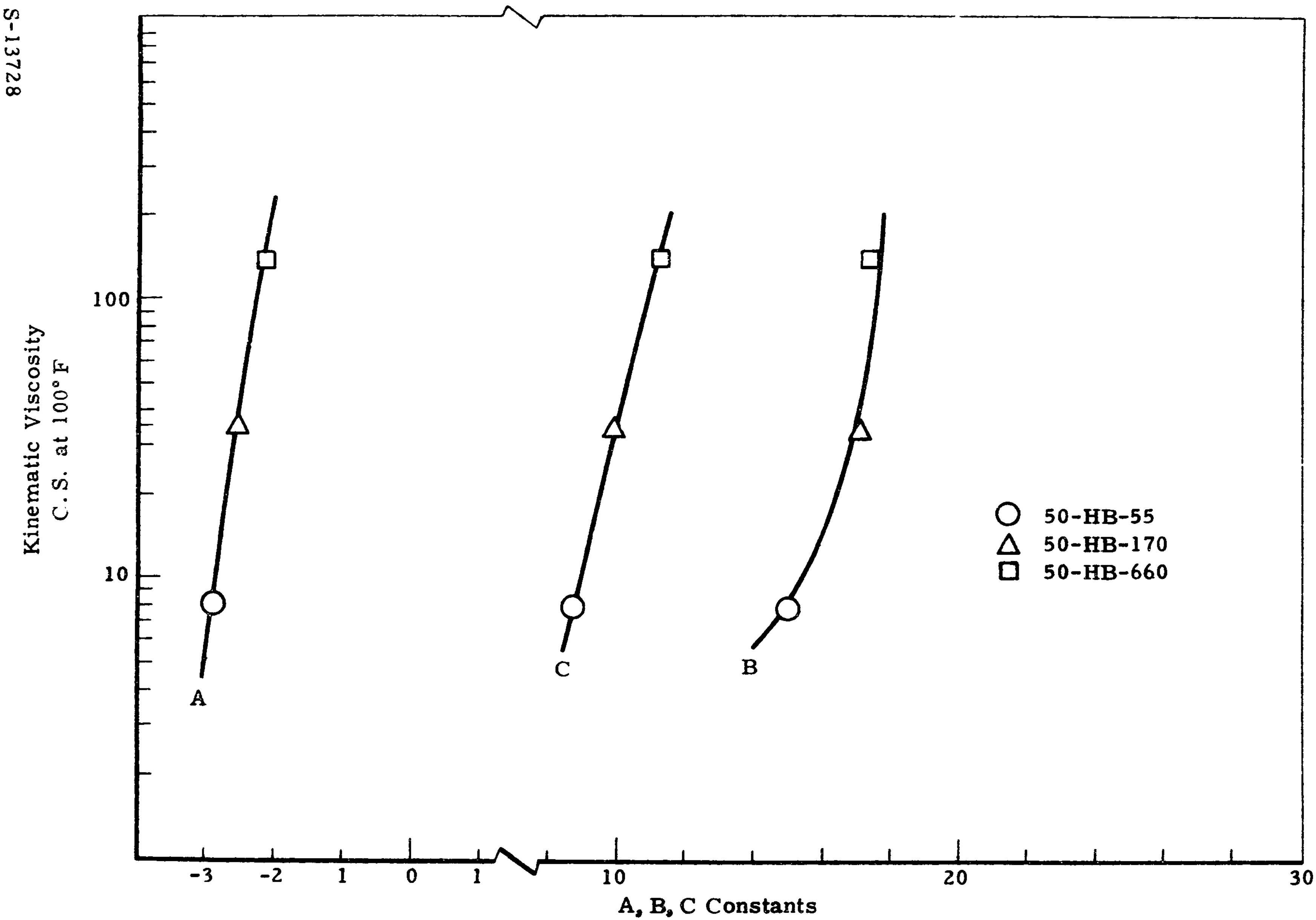

Figure 23. RELATION BETWEEN VISCOSITY AND CONSTANTS A, B AND C FOR UCON 50-HB

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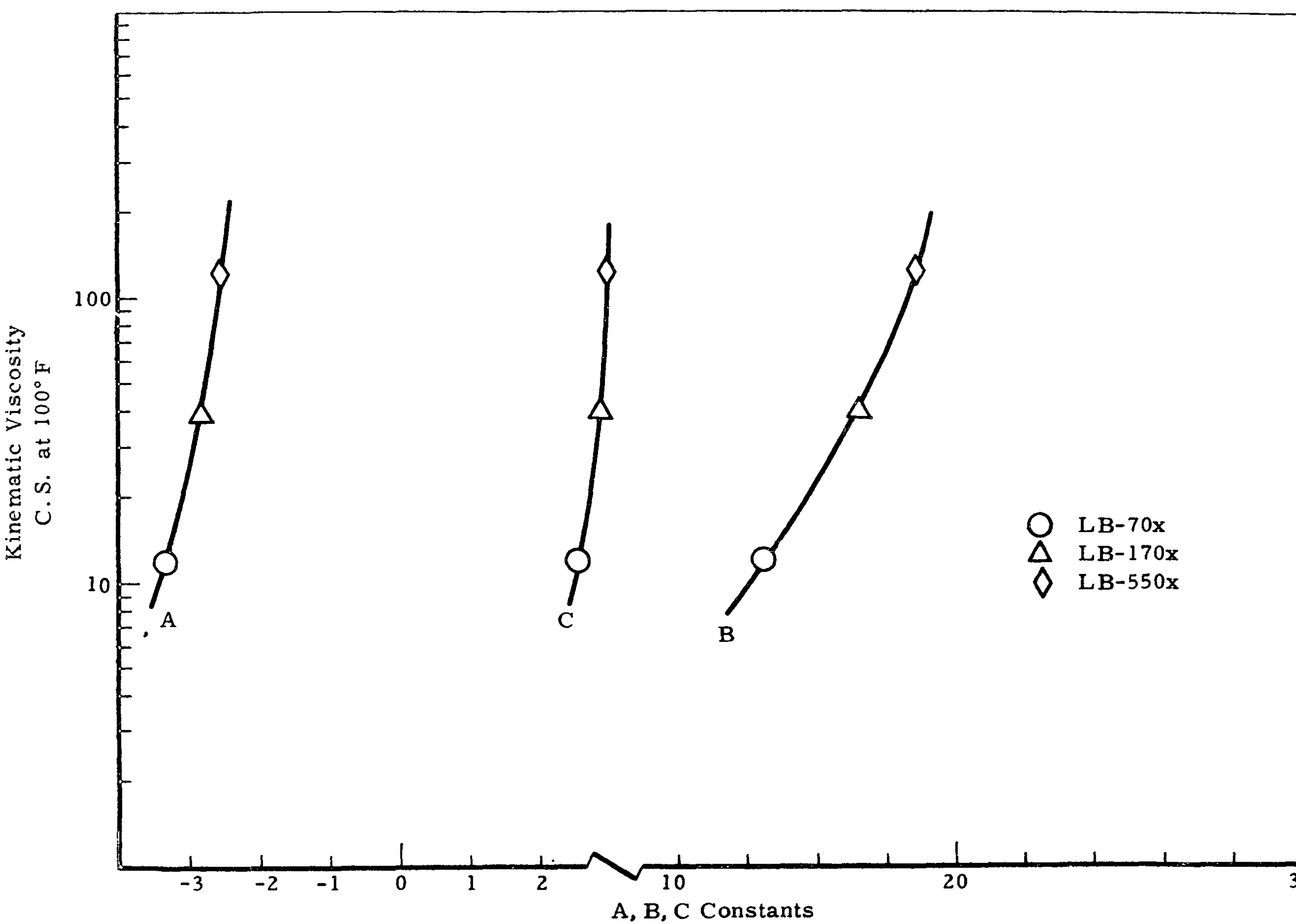

Figure 24. RELATION BETWEEN VISCOSITY AND CONSTANTS A, B AND C FOR UCON LB

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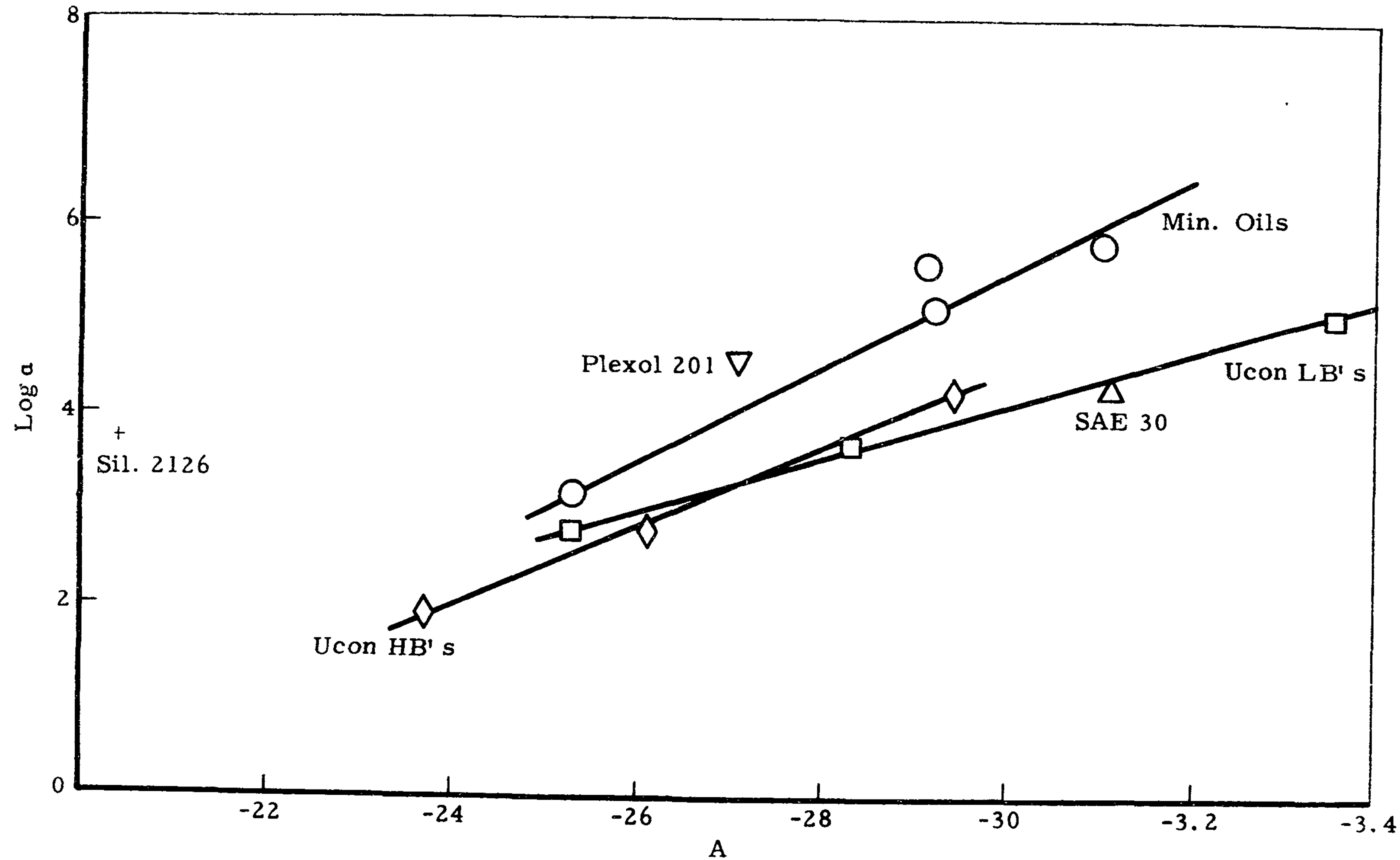

Figure 25. RELATION BETWEEN LOG α AND CONSTANT A

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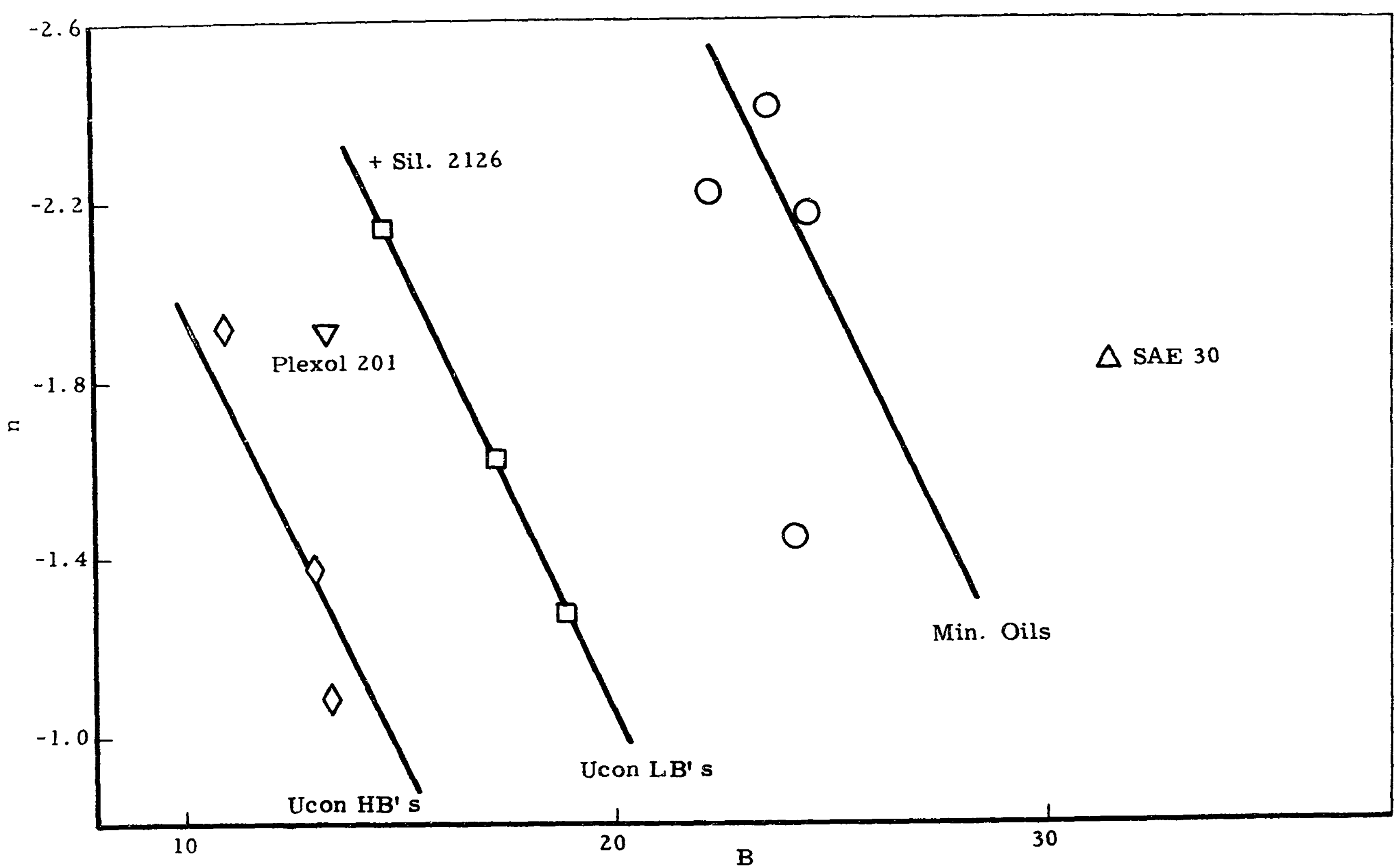

Figure 26. RELATION BETWEEN n AND CONSTANT B

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The Effect of Hardness on Gear Performance

Test Equipment and Procedure. With the exception of a certain type of worm gear drive most of the power transmission gears are manufactured of tough and hard steels. This practice is not a new one and was arrived at through long experience. However, no systematic data on the effect of hardness on gear performance could be found in the literature and for this reason an investigation of the effect of hardness was undertaken. A special set of test gears, similar to our standard test gears, was procured. They were 17 and 19 tooth gears of six diametral pitch, 20° pressure angle and .25 in face width, having a surface finish of 20 microinches. These gears were divided into six groups of 10 pairs, which had been hardened to 30, 35, 40, 45, 50 and 55 Rc. To complete the series our regular test gears of 62 Rc hardness were used. The gears were inspected for eccentricity, spacing and involute errors, surface finish, and hardness. The results are given in Table 7. As can be noted the combined accumulated errors are high. A somewhat poorer accuracy of results was, therefore, expected, and the conclusions will be based not on the results of individual tests but on the general trend of the data.

The tests consisted of determining the load carrying capacities of oils at different speeds using the gears of different hardness. The oils used were 1010 grade, SAE 30 and SAE 60 mineral oils, and SAE 30 mineral oil compounded with 0.5%w S in the form of dibenzyl disulfide. These tests were performed at 3200, 5000, 10,000 and 15,000 rpm gear speeds with a lubricant supply temperature of 100°F and a flow-rate of 10 ml/sec. The load was increased by 4 lb steps, running 5 minutes at each load setting.

The results of these tests are presented in Table 8 and also by the photographs of gear working surfaces given in Figure 27 and Figure 28.

Discussion of Results. The tests disclosed that soft gears are subject to failures by pitting and plastic deformation and that these failures are more susceptible to changes in hardness than are failures by scoring. Pitting and plastic deformation are therefore included in Table 8. Since no units for these failures have been assigned they are simply described as light, medium and heavy at the score loads of these tests. The data given in Table 8 represent the average values of the results. The tests at each set of operating conditions were performed in duplicate when the values of score load obtained checked within 4 lb beam load. In a few cases the score loads differed by 8 lb, and in one case the difference was 12 lb. In these cases the tests were repeated for the third time. The reproducibility in general was poorer than usual due to the greater inaccuracies in gear construction.

The general trend of the effect of hardness appeared to be as follows: At slow speeds soft gears scored at somewhat higher loads than hard gears. This was explained by the higher ability of the soft gears to accomodate plastically, thus uniformly distributing the applied load. At high speeds hard gears carried higher loads. Evidently at high speeds plastic deformation of soft metal was accompanied by temperature increases sufficiently high to promote scoring. In all cases, however, the intensities of scoring with softer steels were larger than with hard steels. The differences in score loads of soft and hard gears in general were not large and from a practical standpoint appeared to be of no importence, because with soft gears such failures as pitting and plastic deformation of the tooth working surface occurred,as a rule, before the score load was reached.

The evaluation of the effect of hardness on pitting and plastic deformation of gears requires a more exact test procedure than used in the tests reported. The time of running at each load should be introduced as an operating variable, and some sort of measure of the intensity of pitting and plastic deformation should be devised. The present tests gave only a general idea of the effect of hardness and permitted only one definite conclusion: The harder are the gears the less are pitting and plastic deformation. The photographs of gear teeth given in Figures 27 and 28 illustrate the superiority of harder gears. The teeth shown belonged to the gears operated at 3200 rpm. As can be noted, in the case of each oil tested the harder were the gears the less pitting and plastic deformation occured; for gears of the same hardness, pitting and plastic deformation increased with load. For example, a gear of 30 Rc hardness when operated with 1010 grade mineral oil scored at 24 lb beam load. Both pitting and plastic deformation were light. A similar gear operated with a heavier SAE 30 mineral oil scored at 38 lb. Pitting was again light, but the plastic deformation of the tooth working surface was deeper. Still heavier plastic deformation may be noted for a gear operated with SAE 60 mineral oil giving a score load of 52 lb. Finally, with the extreme pressure compounded oil the score load was 60 lb, and both pitting and plastic deformation were heavy. It is of interest that the hardest gear (Rc 62) used in no case gave any sign of plastic deformtion or pitting.

Conclusions. (1) The effect of hardness on load carrying capacities of oils is slight. At slow speeds softer gears scored at somewhat higher loads. At high speeds, harder gears withstand heavier loads. In general, the intensity of scoring of soft gears is heavier than that of hard gears. (2) The effect of hardness on scoring is of no practical importance, because soft gears, as a rule, pit and became plastically deformed before reaching the score load. (3) With respect to pitting and plastic deformation hard gears are definitely superior to soft gears.

Table 7. INSPECTION OF GEARS OF DIFFERENT HARDNESS

Nominal Hardness Rc	Gear	Inspections				
		Hardness Rc	Eccentricity (max) in.	Spacing Error (max) in.	Involute Error (max) in.	Surface Finish microinches
30	17	27.5 to 28	0.0003	0.0004	0.0002	22 to 32
	19	27.5 to 29	.0002	.0004	.0002	22 to 30
35	17	35 to 36.5	.0005	.0004	.0003	22 to 48
	19	35 to 36	.0003	.0003	.0003	14 to 26
40	17	39 to 40	.0003	.0003	.0002	12 to 18
	19	38.5 to 39.5	.0004	.0003	.0004	20 to 30
45	17	45 to 45.5	.0002	.0003	.0002	12 to 18
	19	44.5 to 45.5	.0002	.0003	.0002	26 to 36
50	17	50.5 to 51.5	.0003	.0002	.0003	24 to 38
	19	50.5 to 51.5	.0005	.0002	.0003	26 to 38
55	17	52 to 52.5	.0004	.0005	.0003	24 to 36
	19	53 to 54	.0005	.0003	.0003	28 to 38
62	17	61 to 62	.0005	.0004	.0002	
	19	61.5 to 63	.0004	.0004	.0002	

Table 8 follows
Figures 27 and 28 follow

Table 8. THE EFFECT OF HARDNESS ON GEAR PERFORMANCE

Lubricant	Hardness Rc	3,[illegible] rpm					5,000 rpm					10,000 rpm					15,000 rpm				
		Scoring					Scoring					Scoring					Scoring				
		Load lb	Severity	Abrasion	Pitting	Plastic Deform.	Load lb	Severity	Abrasion	Pitting	Plastic Deform.	Load lb	Severity	Abrasion	Pitting	Plastic Deform.	Load lb	Severity	Abrasion	Pitting	Plastic Deform.
SAE [illegible] Min. Oil	[illegible]	[illegible]	Heavy	Medium	Light	Medium	[illegible]	Medium	Medium	Light	Medium	32	Medium	Medium	Light	Light	22	Medium	Medium	None	Light
" "	[illegible]	[illegible]	Heavy	Medium	Light	Light	[illegible]	Medium	Medium	Light	Light	32	Medium	Medium	Light	Light	24	Medium	Light	None	Light
" "	[illegible]	[illegible]	Heavy	Medium	Light	Light	[illegible]	Medium	Light	Trace	V. Light	[illegible]	Medium	Light	None	V. Light	24	Medium	Light	None	V. Light
" "	[illegible]	[illegible]	Heavy	Medium	V. Light	V. Light	[illegible]	Medium	Light	Trace	Trace	30	Medium	Light	Trace	Trace	28	Medium	V. Light	Trace	Trace
" "	[illegible]	[illegible]	Medium	Medium	Trace	None	[illegible]	Medium	Medium	None	None	22	Medium	Medium	None	None	26	Medium	Medium	None	None
" "	[illegible]	[illegible]	Medium	Medium	None	None	[illegible]	Medium	Medium	None	None	28	Medium	Medium	None	None	26	Medium	Medium	None	None
" "	[illegible]	[illegible]	Medium	Medium	None	None	[illegible]	Medium	Medium	None	None	20	Medium	Medium	None	None	24	Medium	Medium	None	None
SAE [illegible] DBLC	[illegible]	[illegible]	V. Light	Medium	V. Heavy	V. Heavy	44	V. Light	Medium	V. Heavy	V. Heavy	32	V. Light	Medium	Heavy	Heavy	32	Light	Light	Heavy	Medium
" " "	[illegible]	[illegible]	V. Light	Medium	V. Heavy	V. Heavy	[illegible]	V. Light	Medium	V. Heavy	Medium	40	V. Light	Light	Heavy	Heavy	24	V. Light	Light	Heavy	Medium
" " "	[illegible]	[illegible]	Light	Medium	Light	None	[illegible]	Medium	Medium	None	None	52	Light	Medium	Medium	None	40	V. Light	Light	Medium	None
" " "	[illegible]	[illegible]	Light	Medium	None	None	[illegible]	Medium	Medium	None	None	[illegible]	Light	Medium	None	None	40	V. Light	Light	V. Light	None
" " "	[illegible]	[illegible]	Light	Medium	None	None	[illegible]	Medium	Medium	None	None	44	Light	Medium	None	None	50	Light	Medium	None	None
SAE [illegible] Min. Oil	[illegible]	[illegible]	Light	Medium	None	Medium	[illegible]	Medium	Medium	V. Light	Medium	32	Medium	Medium	None	Light	36	Light	Light	Heavy	Light
" "	[illegible]	64	Light	Medium	Heavy	Light?	[illegible]	Light	Medium	None	None	56	Light	Medium	None	None	32	Light	Light	None	None
" "	[illegible]	[illegible]	Light	Medium	None	None	[illegible]	Light	Medium	None	None	40	Light	Medium	None	None	32	Light	Light	None	None
[illegible] Grade Min. Oil	30	24	Heavy	Medium	Light	V. Light	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
" " " "	35	24	Heavy	Medium	Light	Light	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
" " " "	40	32	Heavy	Medium	Light	V. Light	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
" " " "	[illegible]	24	Medium	Medium	None	None	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-
" " " "	[illegible]	[illegible]	Medium	Medium	None	None	-	-	-	-	-	-	-	-	-	-	-	-	-	-	-

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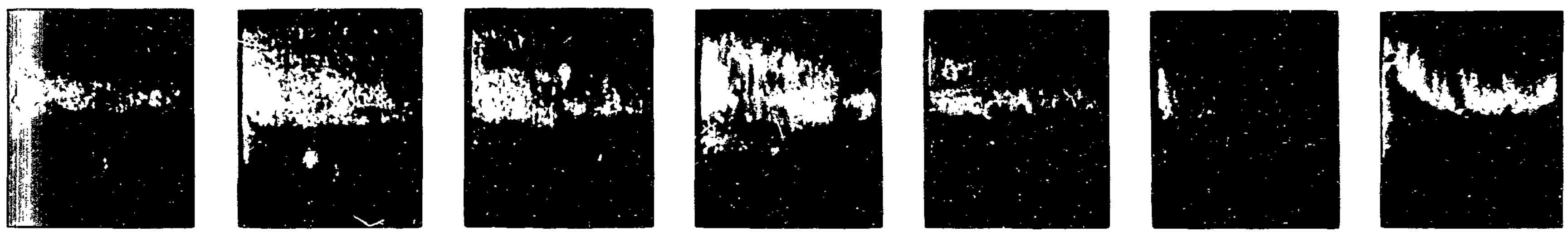
SAE 30 Mineral Oil



30 35 40 45 50 55 62

Hardness Rc

SAE 30 Mineral Oil + 0.5% S Dibenzyl Disulfide

30 35 50 55 62

Hardness Rc

Figure 27. EFFECT OF HARDNESS GEAR TEETH AFTER OPERATION AT 3200 RPM

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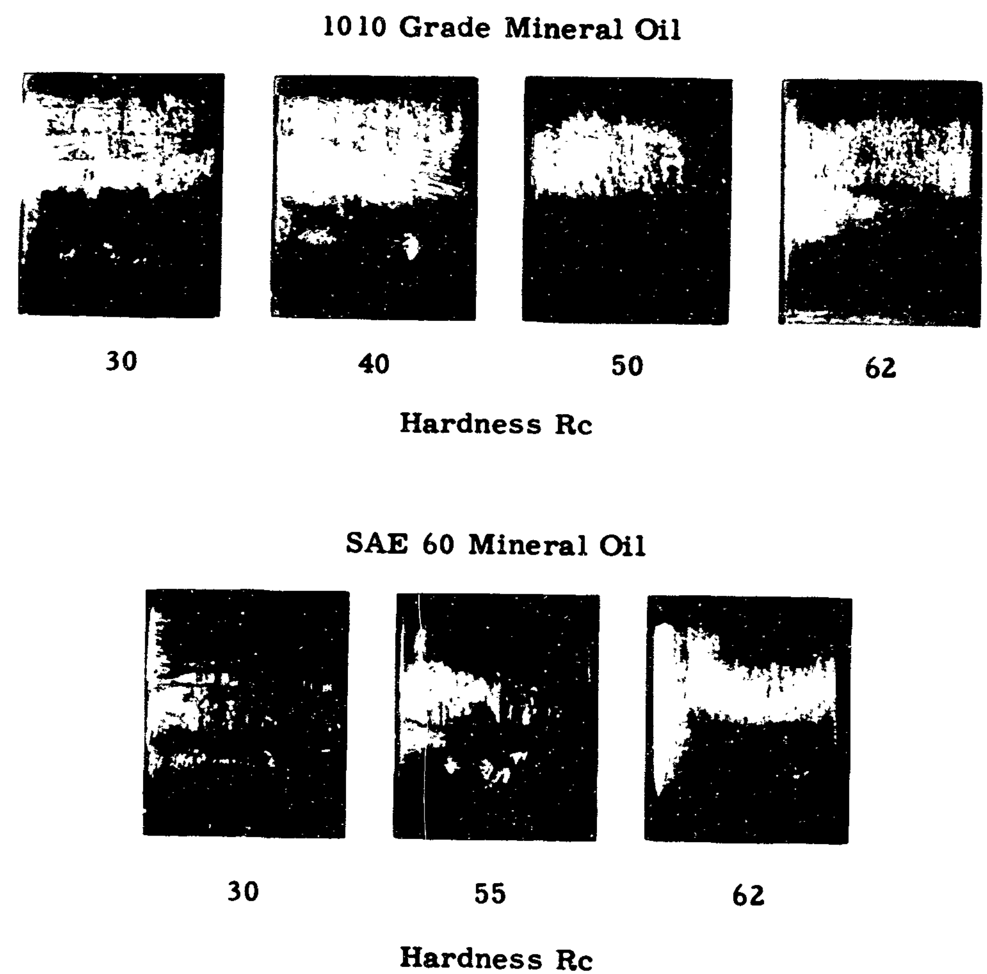

Figure 28. EFFECT OF HARDNESS GEAR TEETH AFTER OPERATION AT 3200 RPM

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The Effect of Gear Face Width on Load Carrying Capacity Ratings

As was mentioned above a quantitative description of the load carrying capacity of lubricating oils in gear systems in relatively fundamental terms is inordinately difficult because of the large number gear geometry and operating variables involved and the inability to measure and control some of these variables. As a compromise, load carrying capacities of oils are often expressed at present in terms of the load per unit of face width of the gear. The gears in actual use, however, vary in their face width from a fraction of an inch to many inches and, indeed, it is important to know how the variations in the face width affect the load carrying capacity ratings. For this purpose a small number of the tests were performed.

The test gears and equipment available limited these investigations to the range between 0.10 and 0.50 inches for the length of contact line. The tests were performed with SAE 30 mineral oil, and the load carrying capacity at different lengths of contact and speeds was established. As in all our load carrying capacity tests, the oil supply temperature was maintained at 100°F, oil flow-rate was 10 ml/sec and load increments were 4 lb beam load, running 5 minutes at each load setting. The lengths of contact used were 0.10, 0.25 and 0.50 in.

The results of these tests are presented in Table 9 and are plotted in Figure 29 and Figure 30.

Discussion of Results. The load carrying capacity of an oil can be expressed in terms of a beam, or total, load carried or as a load per unit of length of contact. The relation of load carrying capacity expressed in terms of the beam load to the width of gear face is shown in Figure 29. This relation is represented by the straight lines the slopes of which are affected by the gear speed. It follows that the wider are the gears the heavier load they carry, a relation which is logical and to be expected.

The relation between unit load carried by gears and their width is presented in Figure 30. Here, the wider are the gears the smaller the unit load they carry. This relation is also affected by speed. The decrease in the unit load with the increase in width of gear faces could be explained by inaccuracies in alignment of the mating teeth and inaccuracies in the faces themselves. These results also show that expressing the load carrying capacities of oils in terms of a unit load does not give an "absolute" measure for the ability of an oil to carry the load. The load carrying capacity of an oil is better described if both length of contact and speed are specified.

Conclusions. (1) Load carrying capacity when expressed as the beam, or total, load increases with the increase in width of gear faces. (2) When expressed as a unit load, load carrying capacity decreases with the increase in width of gear faces. (3) For a better description of load carrying capacity of oils the length of tooth contact and speed used in testing should be specified.

Table 9. EFFECT OF GEAR FACE WIDTH ON LOAD CARRYING CAPACITY OF SAE 30 MINERAL OIL

Speed, rpm	Load Carrying Capacity at Various Width of Contact					
	0.10 in.		0.25 in.		0.50 in.	
	Beam Load, lb	Unit Load, lb/in	Beam Load, lb	Unit Load, lb/in	Beam Load, lb	Unit Load, lb/in
1,000	36	6370	50	3540	76	2690
3,200	16	2830	24	1700	36	1270
5,000	12	2120	16	1130	24	840
10,000	16	2830	20	1420	28	990
15,000	16	2830	24	1700	-	-
20,000	20	3540	24	1700	-	-
25,000	24	4250	-	-	-	-
30,000	OK at 28	4960	-	-	-	-

Figures 29 and 30 follow

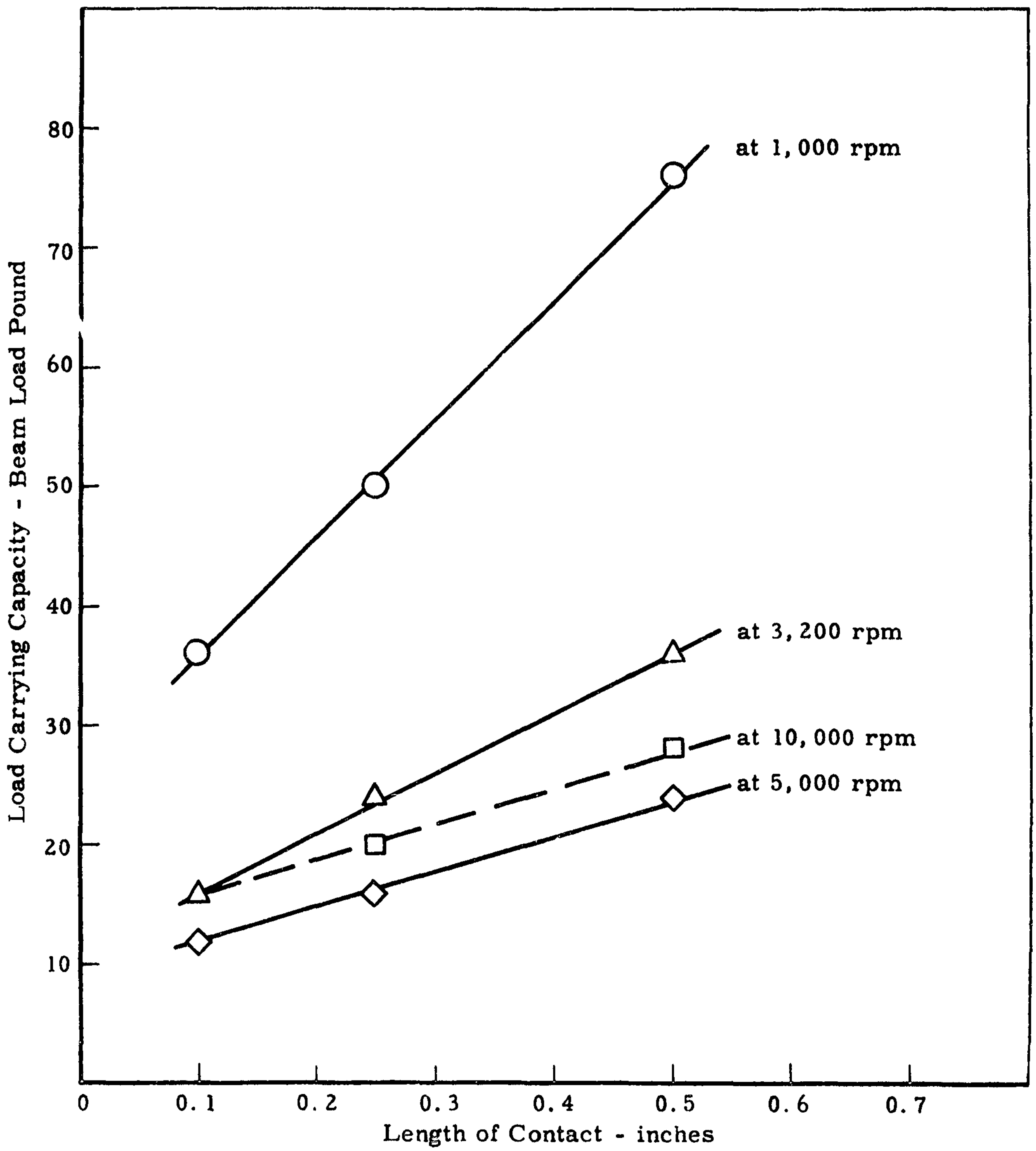

Figure 29. THE RELATION BETWEEN LENGTH OF CONTACT AND SCORE LOAD

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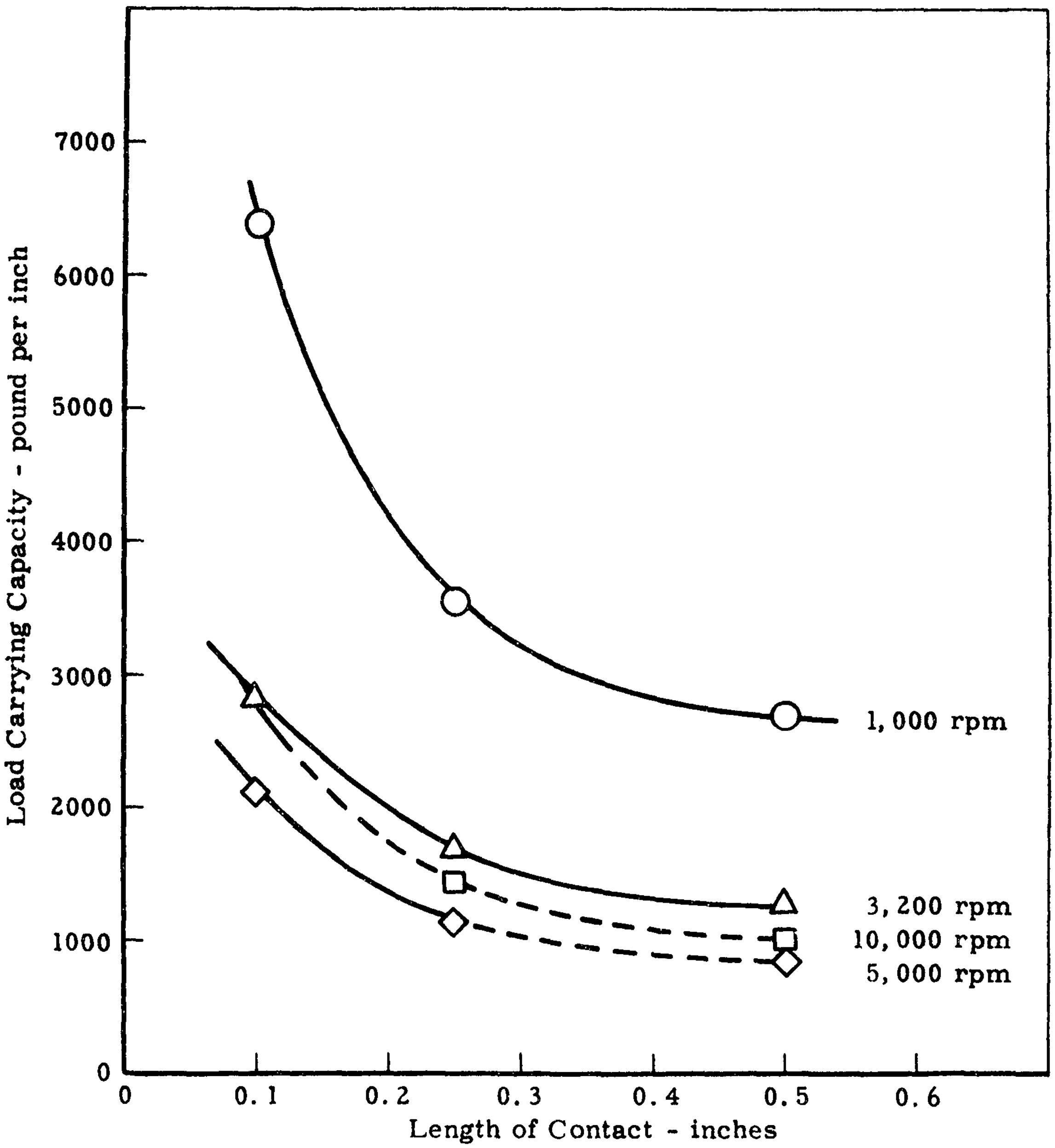

Figure 30. THE RELATION BETWEEN LENGTH OF CONTACT AND SCORE LOAD EXPRESSED AS A UNIT LOAD

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Load Carrying Capacity of Gear Oils at High Speeds

The effect of speed on load carrying capacity of oils was one of the first items studies under this Contract. The speed range between 1,000 and 20,000 rpm was investigated. The results of these tests were reported in the bimonthly progress report No. 2 and also Summary Report S-13605. During the second year of these investigations additional data were secured for silicones and UCON oils of various viscosities. These results were presented in Summary Report S-13649. All these data showed that the load carrying capacities of unreactive oils decreased with the increase in speed in the speed range between 1,000 and 10,000 rpm and that in the higher speed range between 10,000 and 20,000 rpm the effect of speed was opposite to that of low speed, i.e. load carrying capacity increased with the increase in speed. This unexpected effect of high speeds was explained as being due to elasto-plastic behavior of liquids when subjected to an extremely rapid deformation pulse. The residence or contact times for the teeth in mesh were calculated for each of the test conditions used; and it was shown that at contact times of the order of 10^{-6} seconds or less, the relaxation phenomenon exercised its influence on load carrying capacities of oils.

Data in still higher speed range were promised, but the tests were withheld to the very end of these investigations, because any breakage of the machine at speeds over 20,000 rpm could be catastrophic and would require a lengthly repair. The studies of scoring performance of oils in the speed range between 20,000 and 30,000 rpm were conducted, however, during the last four months of the investigations under this Contract.

The tests were performed in the High Speed Spur Gear Machine using our standard 17-19 tooth test gears. Tests conditions were the same as used in previous investigations on the effect of speed, viz. a flow rate of lubricant of 10 ml/sec, on oil supply temperature of 100°F, and load increments of 4 lb beam load, running 5 minutes at each load setting. At the end of each running period the machine was stopped and the working surfaces of the teeth were microscopically inspected.

Before proceeding with regular testing a series of exploratory runs at high speeds were conducted to make sure of the good mechanical condition of the machine and to familiarize the operator with making observations at the high noise level. During these runs it was noted that load carrying capacities of oils at high speeds are generally very high, and to avoid a heavy loading of the machine the regular tests were performed with gears off-set to give 0.1 in length of contact. With gears offset in this way 1 lb of beam load corresponds to 177 pounds per inch of face, and a load increment of 4 lb is theoretically equivalent to 10 lb for gears with 0.25 in contact length. In Table 10 and also in Figures 31 and 32 the values of score loads obtained with the gears offset to 0.1 in were given in terms of standard 0.25 in contact. The convertion was made strictly on the basis of proportionality of the lengths of contact. However, in view of the findings presented in the previous chapter (page 17), this proportionality does not hold true for score loads and the values given are approximate. No corrections for this effect were made, because the data on the effect of face width at speed between 20,000 and 30,000 rpm were not yet obtained.

The results of the tests on the effect of speed at 25,000 and 30,000 rpm together with that of the previous tests are presented in Table 10 and are plotted in Figure 31 and Figure 32.

Discussion of Results. During the tests at 25,000 and 30,000 rpm no troubles of any kind were experienced with the machine itself. However, the Gilmer toothed belts used for driving the machine lasted for only a few tests. A new supply of these belts was ordered and received on time, but the belts of the new batch proved to be inferior, lasting for only a few minutes at 30,000 rpm. The manufacturer of these belts, the New York Packing and Belting Company, was approached and we were informed that a new and improved type of belts is in process of development and would soon be available. In the meantime, while our supply of the belts lasted, we were able to perform only a limited number of the proposed tests.

The data on the effect of speed on load carrying capacity of oils are given in Table 10. Here, the data for the speed range between 1,000 and 20,000 rpm were obtained previously and were reported and discussed in our bimonthly progress report No. 2 and Summary Report S-13605. The data for speeds of 25,000 rpm and 30,000 rpm are new. At these high speeds three mineral oils, one di-ester oil, and one silicone oil were tested. All these oils showed high increases in load carrying capacity with the increase in speed in this high speed range. In fact with SAE 30 and SAE 60 mineral oils and Plexol 201 no score load was reached at 30,000 rpm. The plots for the load carrying capacity - speed relations for mineral oils are given in Figure 31 and for Plexol 201 and Silicone DC-200 AA 2126 in Figure 32.

The fact that all of the oils tested consistently showed high increases in load carrying capacities at speeds over 20,000 rpm indicates that the phenomenon noted is a real one. It appeared that the explanations for the increases of load carrying capacities of unreactive oils with the increase in speed in the high speed zone given previously (see page 12 Summary Report S-13605) could be applied for this still higher speed range. However, it is recognized that the data in the speed range between 20,000 rpm and 30,000 rpm require verification and also should be expanded.

Conclusions. Unusually high increases in load carrying capacities of unreactive oils with the increase in speed in the speed range between 20,000 and 30,000 rpm were observed. The phenomenon noted appeared to be a real one and tentively was explained by the elasto - plastic behavior of liquids at very rapid deformation pulses.

Table 10. EFFECT OF SPEED ON LOAD CARRYING CAPACITY OF OILS

Lubricants	Speed		Width of Contact in.	Score Load			Hertz Stress psi	HP at Scoring
				Beam, lb		Load Per Inch of Face, lb		
	rpm	fpm		For .1 in. Face	For .25 in. Face			
1010 grade oil	1,100	816	.25	-	35	2,520	290×10^3	14.7
	3,200	2,370	.25	-	12	864	163 x "	14.6
	4,800	3,560	.25	-	4	288	98 x "	7.3
	10,000	7,415	.25	-	4	283	98 x "	15.0
	15,000	11,120	.25	-	8	566	139 x "	45.0
	20,000	14,830	.25	-	10	708	155 x "	75.0
	25,000	18,535	.25	-	16	1,132	196 x "	149.7
	30,000	22,245	.10	24	60[a]	4,248	380 x "	675.0[a]
SAE 30 min oil	1,100	816	.25	-	50	3,600	346×10^3	21.0
	3,000	2,220	.25	-	25	1,800	245 x "	28.6
	5,200	3,850	.25	-	15	1,080	190 x "	29.7
	10,000	7,415	.25	-	20	1,416	219 x "	75.0
	15,000	11,120	.25	-	24	1,693	240 x "	135.0
	20,000	14,830	.25	-	24	1,693	240 x "	180.0
	25,000	18,535	.10	24	60[a]	4,248	380 x "	562.5[a]
	30,000	22,245	.10	OK at 28	-	-	-	-
SAE 60 min oil	1,100	816	.25	-	70	5,400	410×10^3	29.4
	3,000	2,220	.25	-	45	3,240	329 x "	51.4
	5,200	3,850	.25	-	35	2,520	290 x "	69.2
	10,000	7,415	.25	-	40	2,832	309 x "	150.0
	15,000	11,120	.25	-	40	2,832	309 x "	225.0
	20,000	14,830	.25	-	40	2,832	309 x "	300.0
	25,000	18,535	.10	32	80[a]	5,664	438 x "	750.0[a]
	30,000	22,245	.10	OK at 28	-	-	-	-
Plexol 201	1,000	742	.25	-	65	4,680	395×10^3	27.2
	3,200	2,370	.25	-	30	2,160	268 x "	36.6
	5,200	3,850	.25	-	15	1,080	190 x "	29.7
	10,000	7,415	.25	-	26	1,842	250 x "	97.2
	15,000	11,120	.25	-	18	1,275	208 x "	101.0
	20,000	14,830	.25	-	24	1,700	240 x "	179.5
	25,000	18,535	.10	20	50[a]	3,540	346 x "	467.5[a]
	30,000	22,245	.10	OK at 28	-	-	-	-
Silicone DC 200 AA 2126	1,000	742	.25	-	35	2,520	290×10^3	14.7
	3,200	2,370	.25	-	8	576	139 x "	9.6
	5,200	3,850	.25	-	4	288	98 x "	7.9
	10,000	7,415	.25	-	8	566	139 x "	29.9
	15,000	11,120	.25	-	14	991	183 x "	78.5
	20,000	14,830	.25	-	10	708	155 x "	74.8
	25,000	18,535	.10	12	30[a]	2,124	268 x "	280.5[a]
	30,000	22,245	.10	20	50[a]	3,540	346 x "	561.0[a]

a) Computed values for 0.25 inch face width from data obtained at 0.1 inch face width.

Figures 31 and 32 follow

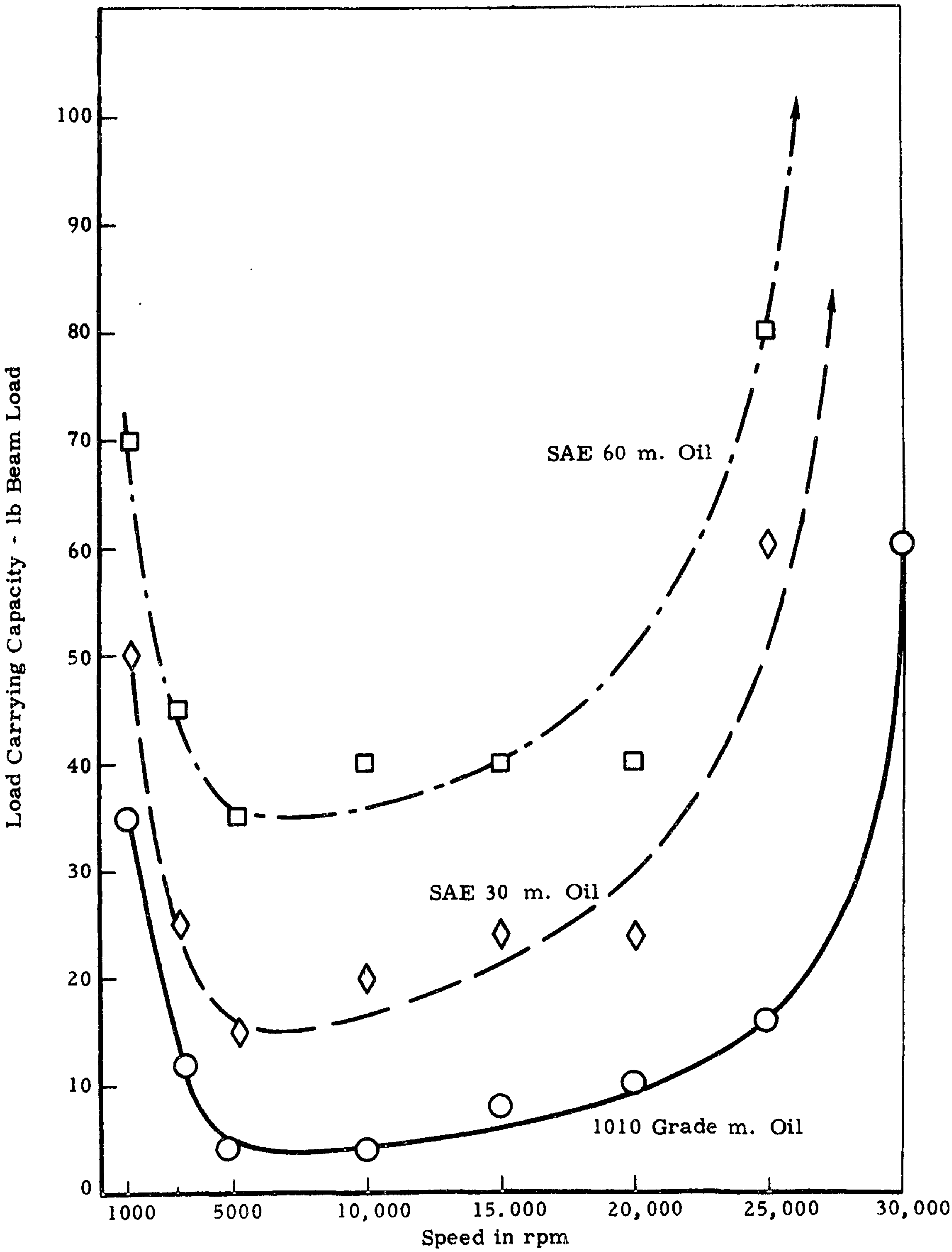

Figure 31. EFFECT OF SPEED ON LOAD CARRYING CAPACITY OF MINERAL OIL

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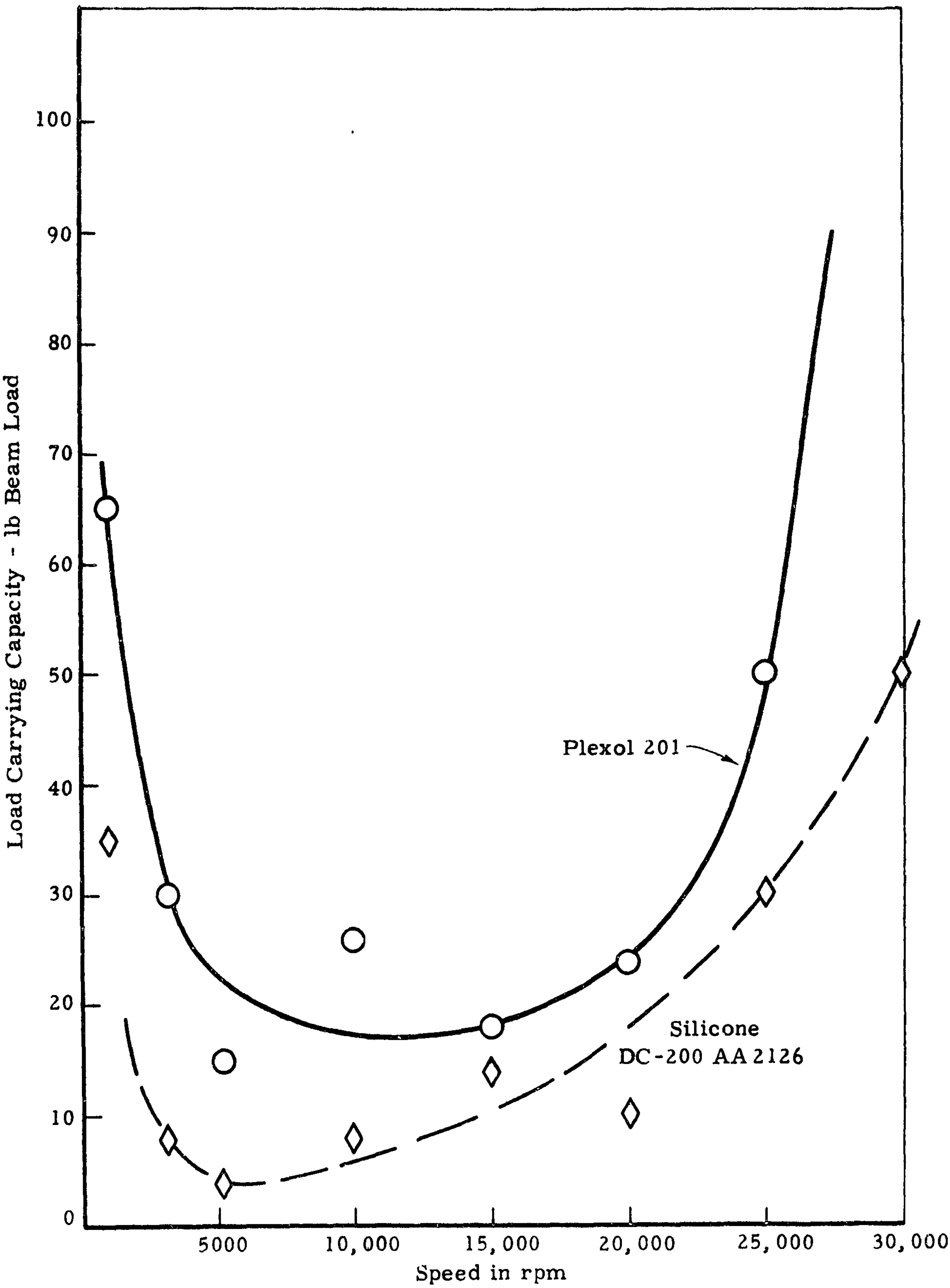

Figure 32. EFFECT OF SPEED ON LOAD CARRYING CAPACITY OF SYNTHETIC OILS

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Performance of Gears and Gear Lubricants at High Temperatures

The ever increasing power and speed of modern planes and rockets subject mechanical parts to increasing temperatures. A knowledge of the behavior of these mechanisms at high temperatures thus becomes most necessary. Since gears are an integral part of many of such mechanisms, studies of gear operations at high temperatures were conducted, for which purpose a special High Temperature Spur Gear Machine was constructed.

Description of High Temperature Spur Gear Machine. The new High Temperature Spur Gear Machine, shown in Figure 33, was constructed along lines similar to those of our oldest Spur Gear Machine described on Page 4, bimonthly progress report No. 1. The schematic diagram of this machine is given in Figure 34. As can be noted, the machine is based on the closed power circuit or four-square principle and essentially consists of two geometrically similar pairs of gears connected by two parallel shafts. One of the shafts is constructed of two sections connected by a special coupling. Loading is accomplished by locking one side of the coupling and applying torgue to the other. The coupling is then bolted together. Each pair of gears operates in a separate gear box. The box on the power transmitting or low temperature side of the machine also contains the supporting bearings. On the test side of the machine the supporting bearings are located in a separate compartment insulated from the hot test chamber by air spacing. The test gear chamber is supported by a stainless steel block insulated at the foundation by a transite lining. This block also contains a pair of auxiliary ball bearings especially designed for high temperature operation (MCS-206-048 Rollway Bearing Co). The test gear chamber is heated by electrical heating elements located in a detachable transite cover built to enclose the gear box snugly. The test oil circulating system consists of a sump, a Zenith Metering Pump No. 5 geared to supply a flow-rate of 10 ml/sec, and a heat exchanger for heating the oils. While in operation at elevated temperatures, the test side of the machine is placed under a hood and the lubricant fumes are drawn out.

The High Temperature Spur Gear Machine is designed to use the same test gears as employed in our older spur gear machines. These gears are 17 and 19 tooth, 6 diametral pitch, 20° pressure angle and 0.25 in face width. The gears are manufactured of SAE 3312 steel (or equivalent) and are case-hardened to 62 Rc with a core hardness of 30 to 40 Rc. The working surfaces are finished to about 20 microinches. The constants for these gears were given in Table 2 and their operatings characteristics in Table 3 of the bimonthly progress report No. 1.

The gear load in this machine is limited to a maximum of 85 lb beam load by the strength of the gears and the strength of the torsion shaft. The speed range is between 1,000 and 10,000 rpm. The capacity of the electrical heaters is 7,000 watts, which permits heating of the test gear compartment to well in excess of 800°F.

Correlation Tests

To establish the correlation in rating of lubricants in the new High Temperature Spur Gear Machine and in our older machines a limited number of oils were tested for load carrying capacity under standardized operating conditions, namely: an oil supply temperature of 100°F, a flow-rate of 10 ml/sec, and load increments of 4 lb, running 5 minutes at each load setting. The results of these tests are presented in Table 11.

As can be noted good correlation was obtained. This is not surprising, because the test gears used were the same and the geometry of the new machine is very similar to the older machines.

Test Procedures and Results. One of the requirements of this Contract calls for the investigation of the high temperature performance of gears in the temperature range of 400°F and 600°F. The term "performance" comprises a combination of various factors. Thus, in the case of high temperature operations it is of interest to learn how high temperatures affect the load carrying capacity of gears and gear lubricants. It should be established whether or not high temperature operation causes some unusual gear failures. Wear of gears operated at high temperatures should be studied and evaluated. Lubricants should be compared for their life and stability in high temperature operation. To investigate these and other performance characteristics not mentioned under all possible operating conditions would require a prohibitive number of tests. These investigations therefore will be limited to evaluation of load carrying capacity of lubricants at 400°F and 600°F operating temperatures. Microscopic examination of gears in the course of our load carrying capacity tests presents an opportunity to study the type of gear failures concurrently. The thermal stabilities of such lubricants as mineral oils, diesters, silicones and UCON fluids were established elsewhere, and it is believed no additional studies of this kind are necessary here.

The temperature range for these investigations was specified simply as 400°F and 600°F. Whether these temperatures should be of the incoming lubricant or of the gears or of the surrounding media was not mentioned. Since in actual service the sources of heat are the machines themselves and the ram action of the air, both of which heat all parts of the mechanism, we decided to use in our investigations the temperature within the gear box as a reference point. This temperature is measured by means of a thermocouple located within the test gear box and is free from the direct effect of the incoming oil. The oil supply temperature during these tests was maintained at 400°F. The tests were performed at a 3,500 rpm pinion speed, an oil flow-rate of 10 ml/sec and load increments of 4 lb beam load, corresponding to 283.4 lb per inch of face. The duration of the runs at each load was extended to 30 minutes for 400°F tests and to 10 minutes for the tests at 600°F.

The procedure used in the tests at 400°F consisted of adjusting the load to the required value, starting the test lubricant circulation, and then gradually increasing the temperature of the lubricant and the temperature within the gear compartment to 400°F. The machine was started after this temperature was reached. It was thought that under such operating conditions the "true" score loads for each lubricant at 400°F could be established. However, it was noted that when the temperature was raised with the gears at stand-still, a certain amount of gummy deposit was formed on the gears, and it was suspected that these deposits were responsible for the unexpected higher load carrying capacities at 400°F.

To check the effect of these deposits a limited number of tests were performed at a 600°F temperature within the gear box using two test procedures. In one the temperature was raised as in the tests at 400°F, with the gears at stand-still; in the other the machine was started under the required load at room temperature, and the temperature was gradually raised while the gears were in operation.

Finally, a limited number of tests on the effect of temperature were performed with SAE 30 mineral oil. The temperatures investigated were 100°F, 200°F, 300°F, 400°F and 600°F. In these tests both the gear compartment and the oil were heated while the machine was in operation.

The results of these tests are presented in Table 12, Table 13 and Table 14.

Discussion of Results. The results of the tests at 400°F are given in Table 12. SAE 20 and SAE 30 mineral oils and Plexol 201 (diethylhexyl sebacate) showed about the same load carrying capacities at 400°F as at 100°F. The other oils scored at somewhat lower loads at 400°F than at 100°F, but the differences were much smaller than expected. In all cases the score marks at 400°F were lighter than at lower temperatures. However, the abrasion marks were more severe. The lighter intensities of scoring at 400°F are tentatively explained by the decrease in hardness of the gear steel and by the presence of gummy deposits on the gear working surfaces. Also, the running time at each load in the 400°F tests was 30 minutes instead of 5 minutes in our regular load carrying capacity tests, and it is possible that the original scoring was more abraded during this longer period. The other data of some interest are those for the three check tests with SAE 30 mineral oil. The reproducibility in these tests was poor and shows the need for the development of a more rigorous test method.

The data for the exploratory tests at 600°F gear box temperature are given in Table 13. The results show that the load carrying capacities of the oils were higher when the temperature was increased with the gears at stand-still than when the temperature was raised with the gears in operation. When the gears were started after reaching 600°F, a considerable amount of gum deposit had formed on the gears, and it appeared that these gum deposits served as protective coating and decreased the tendency of the gears to score. When the temperature was increased with the gears in operation, scoring most probably occured even before reaching a 600°F temperature. In the case of SAE 60 mineral oil the score load was the same for both procedures. As previously established, the load carrying capacity of this oil at 248°F and 3,200 rpm was 10 lb (Table 15, Summary Report S-13605), and at 322°F and 10,000 rpm it was 20 lb (Table 13, Summary Report S-13649). It is conceivable, therefore, that this oil could withstand without scoring the first load of 4 pounds at temperatures around 400°F. Gum deposits, which start to form at around 400°F, could further decrease the scoring tendency of the gears.

The results of the load carrying capacity tests at various temperatures performed with SAE 30 mineral oil are given in Table 14. In these tests the temperature was increased while the gears were in operation. These results showed that the score loads decreased with the increase in temperature up to 300°F. At 400°F and 600°F scoring occured at the application of the first load of 4 lb, the same as at 300°F. It appeared, therefore, that in the tests at 400°F and 600°F scoring actually occured during the heating period and prior to reaching the specified test temperatures.

Conclusions. The experiments described above were exploratory in nature. Consequently, the conclusions drawn should be considered as incomplete and tentative.

1. Due to the thermal instability of the lubricants the operation of gears for any prolonged period of time at 600°F or higher is inadvisable.

2. No new or unusual types of failures were observed during the operations of gears at 400°F and 600°F. Just as at normal operating temperatures, scoring and abrasion were the two primary destructive failures.

3. The load carrying capacities of oils decreased with the increase in temperature. However, at temperatures over 400°F gum deposits are formed by oils. These deposits serve as a protective coating and improve the scoring performances of gears and gear lubricants.

4. Gears manufactured of SAE 3312, or equivalent, steel may suffer some decrease in hardness when subjected to prolonged heating at temperatures over 400°F.

5. A rigorous test procedure for studying performances of oils at high temperatures should be developed before making any attempts to compare performances of different lubricants. Particular attention should be given to the method of heating.

Table 11. CORRELATION TESTS FOR HIGH TEMPERATURE SPUR GEAR MACHINE

Lubricants	Old Spur Gear Machine at 3200 rpm			High Temperature Machine at 3500 rpm		
	At Scoring			At Scoring		
	Beam Load lb	Total Load lb	Load Per Inch of Face	Beam Load lb	Total Load lb	Load Per Inch of Face
1010 grade mineral oil	12	216	864	8	142	568
SAE 20 mineral oil	12	216	864	12	216	864
SAE 30 mineral oil	25	450	1800	20	360	1440
SAE 60 mineral oil	45	810	3240	48[a)]	850	3400

a) Very light scoring

Table 12. SCORING PERFORMANCE OF OILS AT 400°F

Lubricants	Load Carrying Capacity at 100°F, a) lb	Results of Tests at 400°F b)			Remarks
		Load Carrying Capacity, lb	Total Time of Operation, min	Decrease in Gear Hardness Rc Number	
1010 grade mineral oil	12	4	30	0	-
SAE 20 mineral oil	12	12	90	4	-
SAE 30 mineral oil	25	24	180	12	-
SAE 30 mineral oil	-	32	240	-	Temperature overshooted to 440°F.
SAE 30 mineral oil	-	4	30	2	Trouble with oil circulation.
SAE 60 mineral oil	45	28	210	7	-
Plexol 201	30	32	240	11	-
UCON LB-170X	28	12	90	1	-
UCON 50-HB-170	28	16	120	7	-
Silicone DC-200 AA 2126	8	4	30	3	-

a) 3200 rpm and 5 min at each load, including zero load.
b) 3500 rpm and 30 min at each load, excluding zero load.

Table 13. SCORING PERFORMANCE OF OILS AT 600°F

Lubricants	Load Carrying Capacity at 100°F,[a] lb	Temperature Raised With Gears at Stand-still		Temperature Raised With Gears Running	
		Load Carrying Capacity, lb	Decrease in Hardness, Rc Number	Load Carrying Capacity, lb	Decrease in Hardness, Rc Number
SAE 20 mineral oil	12	12	7	4	5
SAE 30 mineral oil	25	12	6	4	4
SAE 60 mineral oil	45	16	6	16	7
Plexol 201	30	20	6	4	4

a) At 3200 rpm.

Table 14. LOAD CARRYING CAPACITY OF SAE 30 MINERAL OIL AT VARIOUS TEMPERATURES

Temperature, °F	Score Load, lb	Changes in Gear Hardness, Rc Number	Remarks
100	20	1	Heavy scoring
200	12	1	Heavy scoring
300	4	2	Heavy scoring
400	>4	3	Light scoring
600	>4	6	Light scoring

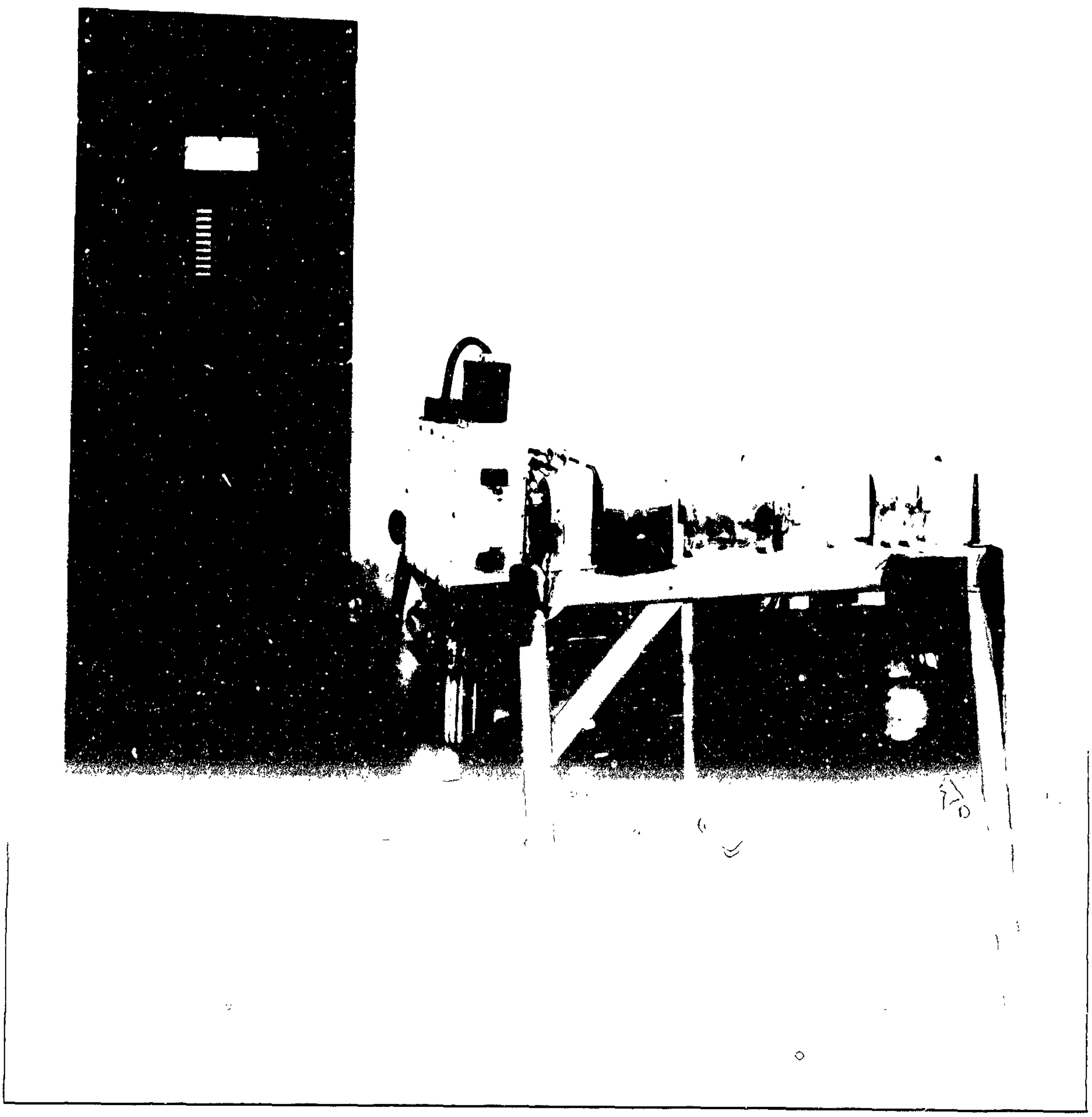

Figure 33. HIGH TEMPERATURE SPUR GEAR MACHINE

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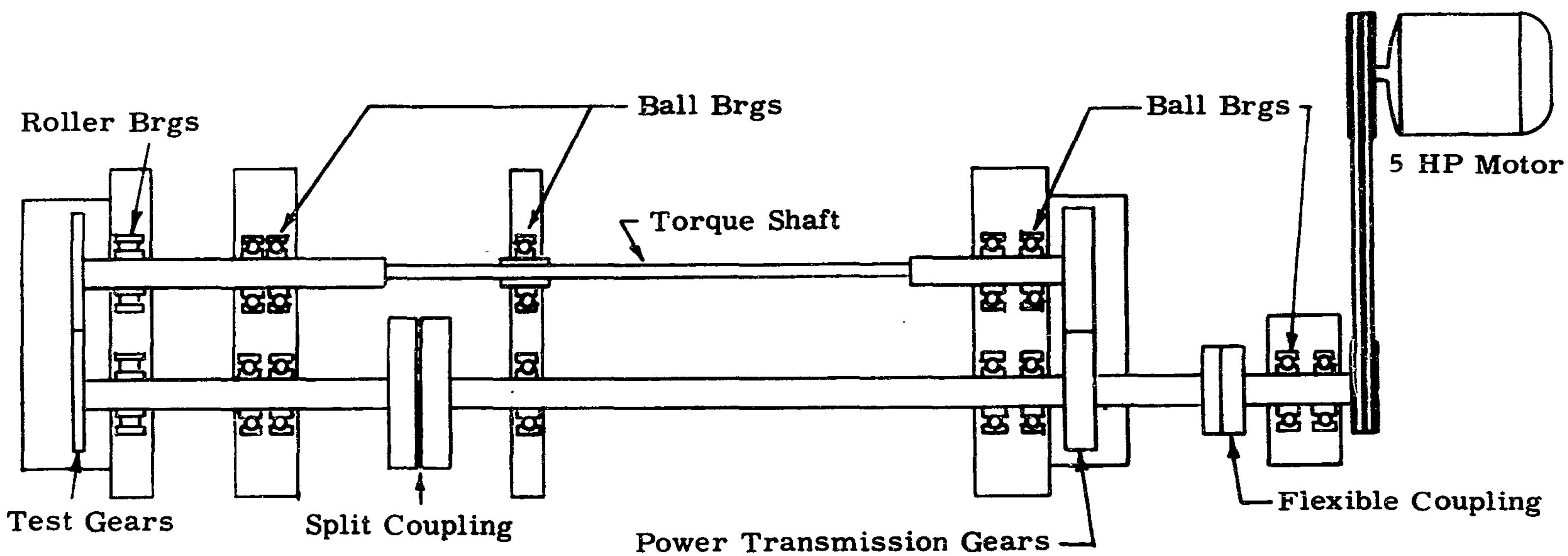

Figure 34. SCHEMATIC DIAGRAM OF HIGH TEMPERATURE SPUR GEAR MACHINE

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The Effect of Speed on Gear Wear

Since reduction, or still better, elimination of wear is one of the most important functions of lubricants, wear was studied extensively throughout these investigations. During the first year of this research it was learned that wear of gears operating below their fatigue limits can be subdivided into: (a) wear by scoring, (b) wear by abrasion, and (c) wear by chemical corrosion. It was found also that unreactive oils give wear mainly through scoring. In operations with extreme pressure compounded oils in the range prior to score loads the mechanisms of wear are abrasion and chemical corrosion. During the second year of these investigations long-term wear was investigated. The results were in full agreement with those of the short duration tests previously performed. It was learned, also, that in operations at loads above the score load of the lubricant the conditions are unstable and any provocations such as shocks or vibrations may cause the occurrence scoring. During the third year the effects of temperature and speed on gear wear were investigated. The tests on the effect of speed were performed with SAE 30 mineral oil and were reported in Summary Report S-13694. The results were of considerable interest, since they suggested the existence of a relation between load carrying capacity of oils and wear. To verify this finding additional tests were made with SAE 60 mineral oil and SAE 30 mineral oil compounded with 0.5%w S in the form of dibenzyl disulfide.

As in the case of the tests with SAE 30 mineral oil these tests were made using our standard 17-19 tooth gears. The 17 tooth pinion gears were irradiated. Wear was measured by placing a scintillation counter in the oil stream, observing the counting rate, and calculating therefrom the amount of metal removed from the radioactive gear. The test conditions were an oil supply temperature of 30°C, a flow-rate of 10 ml/sec, and load increments of 4 lb. The duration of running at each load was 30 minutes. The speeds were 1070, 3200, 5200, 7200, 10,000 and 15,000 rpm. A fresh set of gears was used for the runs at each speed. Load carrying capacities of the lubricants tested were determined by our regular load carrying capacity test using the inert test gears.

The results of these tests are presented in Table 14. The plots of wear and load carrying capacity as a function of speed for SAE 60 mineral oil are given in Figure 35 and for SAE 30 mineral oil compounded with dibenzyl disulfide in Figure 36. The wear at 1070 rpm as a function of load is presented in Figure 37.

Discussion of Results. As shown in Figure 35 the load carrying capacity of SAE 60 mineral oil decreased with the increase in speed reaching a minimum at 5200 rpm. With further increase in speed the load carrying capacity increased. The wear curves show that wear at each speed increased with the increase in load. Comparison of wear at each load for different speeds show that maximum wear occurred at 5200 rpm, when the load carrying capacity was the smallest and that, in general, wear is inversely related to scoring. This relation of wear to load carrying capacity is even more clear in the case of SAE 30 mineral oil compounded with dibenzyl disulfide. As shown in Figure 36 the load carrying capacity of this oil was the smallest at 5200 rpm. It increased when the speed was increased to 10,000 rpm and again decreased at 15,000 rpm. The wear curves followed the reverse pattern of the load carrying capacity curve. These data, therefore, positively confirmed the results previously obtained with the uncompounded SAE 30 mineral oil and suggest the existence of a relation between wear and load carrying capacity of the oils. However, comparision of the wear obtained with the three oils tested suggests that this relation can not be extended from one lubricant to another. As an example, wear obtained at 1070 rpm with the three oils tested was plotted as a function of load in Figure 37. It can be noted that SAE 30 mineral oil gave the smallest wear at loads below 32 lb, although its load carrying capacity was smaller than that of SAE 60 mineral oil or of E.P. compounded SAE 30 oil. Here, it should be mentioned that the tests with SAE 30 oil were performed with the gears free of rust, while the latest tests with SAE 60 mineral oil and SAE 30 mineral oil compounded with dibenzyl disulfide were performed using the irradiated gears of the 1956 batch, which were received somewhat rusted (see bimonthly progress report No. 21). It is possible that somewhat higher wear with the last two lubricants tested could be attributed to the rusted conditon of the irradiated gears.

Conclusions. The conslusions drawn from the above described tests in general confirm the conclusions drawn from the last year experiments (see page 8, Summary Report S-13694). (1) At each speed wear increased with load. (2) With unreactive mineral oils wear is inversely related to load carrying capacity of oils. (3) For oils compounded with dibenzyl disulfide the main mechanism of wear is scoring, as in the case of unreactive oils. (4) The data on the relation between load carrying capacities of different oils and wear should be expanded and verified before the load carrying capacity, as such, could be used as a measure of wear.

Table 15. LOAD CARRYING CAPACITY AND WEAR AT DIFFERENT SPEEDS

SAE 60 Mineral Oil

Speed	Score	Beam Load - lb																
rpm	Load lb	0	4	8	12	16	20	24	28	32	36	40	44	48	52	56	60	64
1070	70	.16	.24	.32	.46	.60	.74	.88	1.10	1.32	1.50	1.68	1.79	9.90	-	9.90	-	10.50
3200	45	.14	.29	.45	.95	1.48	2.30	3.10	excessive wear									
5200	35	.38	.56	.81	.94	1.30	4.82	7.55	9.11	excessive wear								
7200	36	.29	.49	.70	.78	1.18	3.66	6.16	11.30	excessive wear								
10000	40	.25	.50	.67	.90	.90	.80	1.01	1.22	1.66	excessive wear							
15000	38	.12	.25	.46	.71	.71	.95	1.30	2.73	3.85	excessive wear							

SAE 30 Mineral Oil + 0.5%w S Dibenzyl Disulfide

Speed rpm	Score Load lb	0	4	8	12	16	20	24	28	32	36	40	44	48	52	56	60	64
1070	72	.25	.38	.52	.68	.84	1.28	1.71	2.10	2.43	2.90	3.28	3.64	4.01	4.90	5.80	7.12	8.45
3200	44	.10	.30	.48	.68	.87	1.88	2.83	7.55	12.22	excessive wear							
5200	36	.74	1.23	1.59	1.92	2.20	3.72	6.90	excessive wear									
7200	36	.64	.91	1.18	1.48	2.05	2.25	3.32	excessive wear									
10000	60	-	.99	1.05	1.44	1.59	1.90	2.20	2.43	2.73	4.72	5.10	5.60	6.05	7.95	excessive wear		
15000	48	-	2.99	3.37	4.10	4.95	6.25	7.65	9.55	11.70	14.05	excessive wear						

SAE 30 Mineral Oil

Speed rpm	Score Load lb	0	4	8	12	16	20	24	28	32	36	40	44	48	52	56	60	64
1070	50	.0	.03	.07	.15	.23	.30	.38	.53	.69	excessive wear							
3200	25	.07	.15	.23	.85	1.45	2.50	2.85	3.80	excessive wear								
5200	15	.23	.46	1.26	3.65	excessive wear												
7000	-	.23	.31	.54	2.20	excessive wear												
10000	20	.23	.72	.45	1.73	2.47	excessive wear											
15000	24	.09	.18	.36	1.17	2.20	excessive wear											

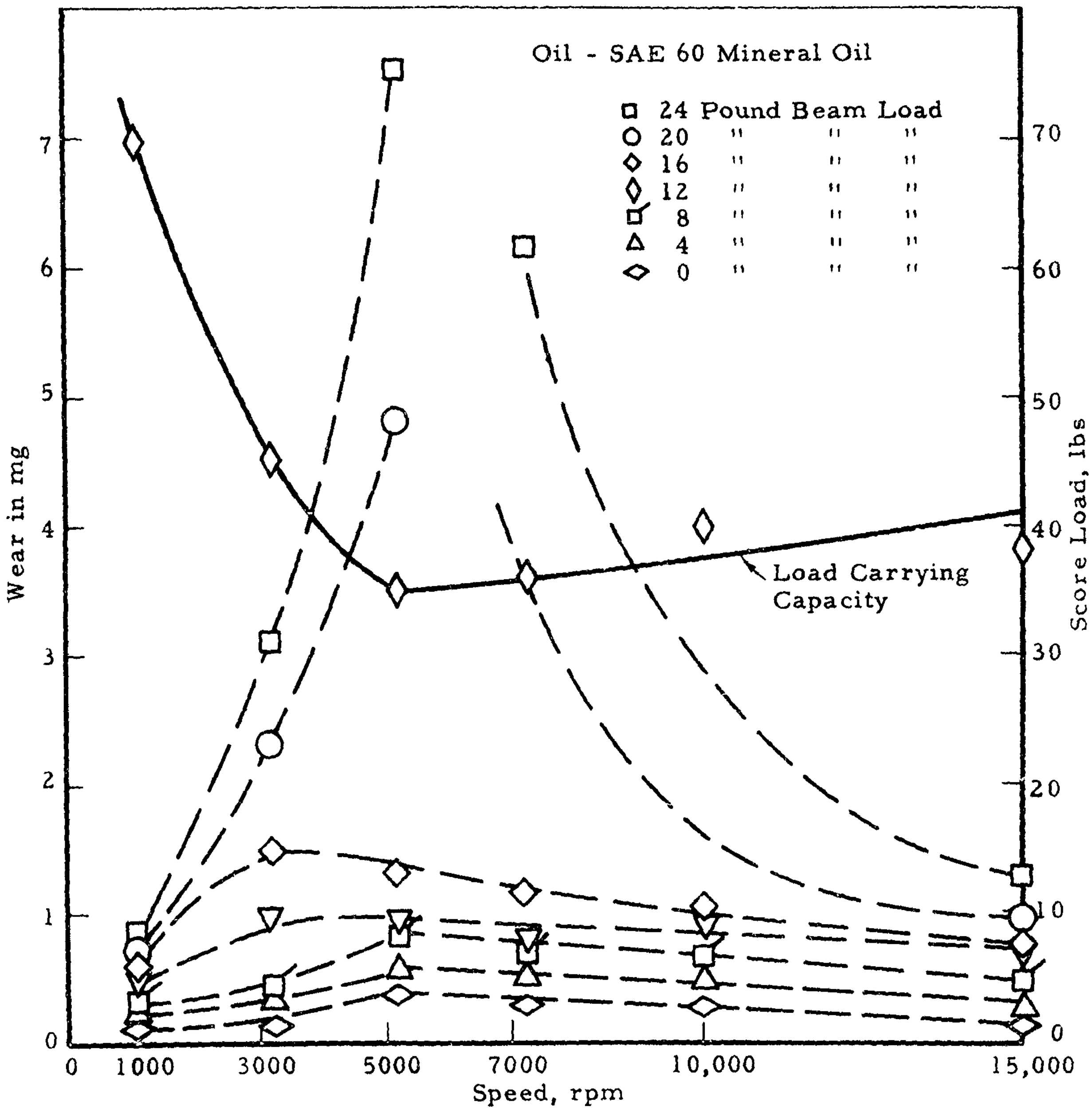

Figure 35. WEAR AND LOAD CARRYING CAPACITY VS SPEED

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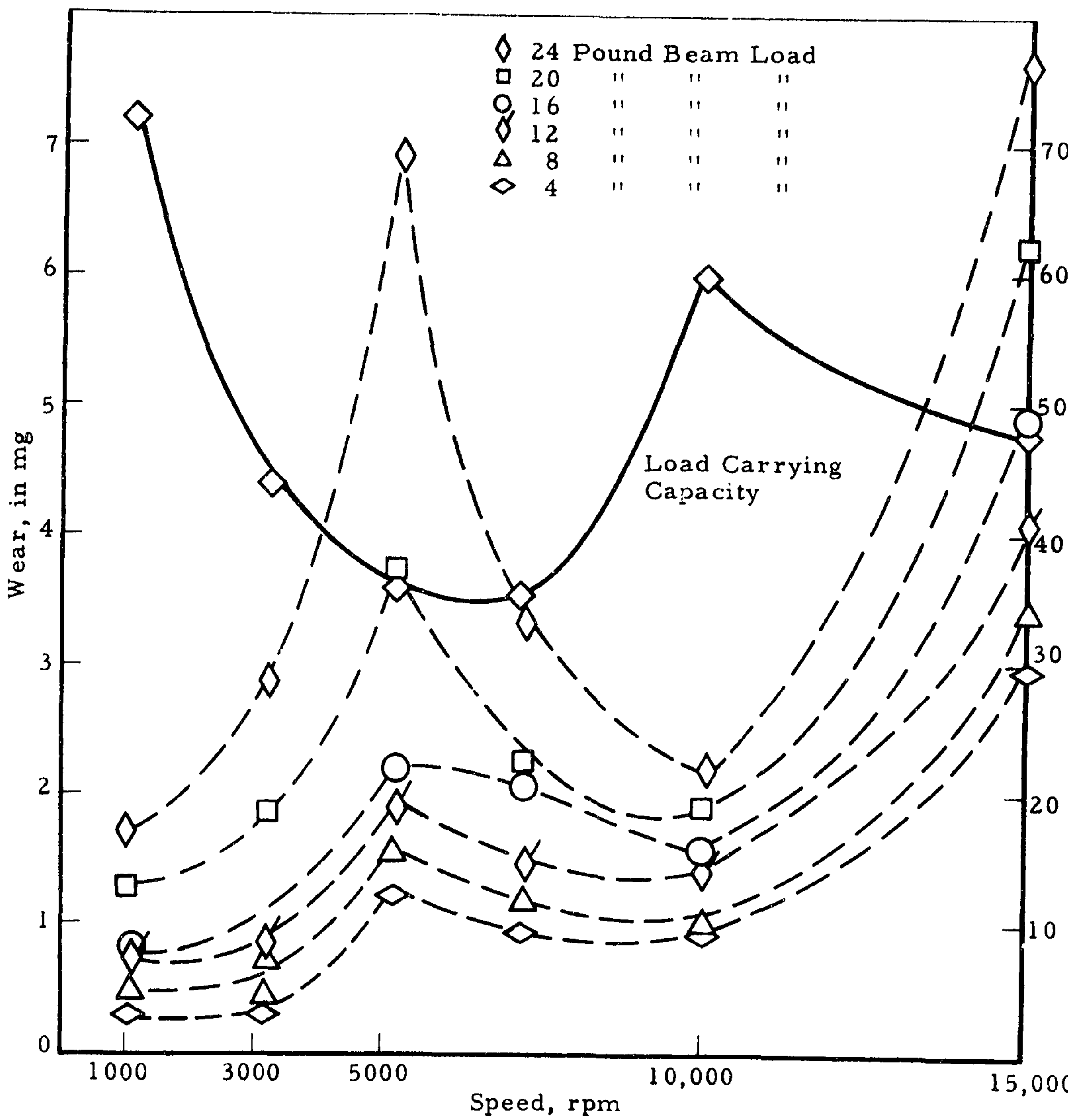

Figure 36. WEAR AND LOAD CARRYING CAPACITY VS SPEED
SAE 30 MINERAL OIL + 0.5%w S DIBENZYL DISULFIDE

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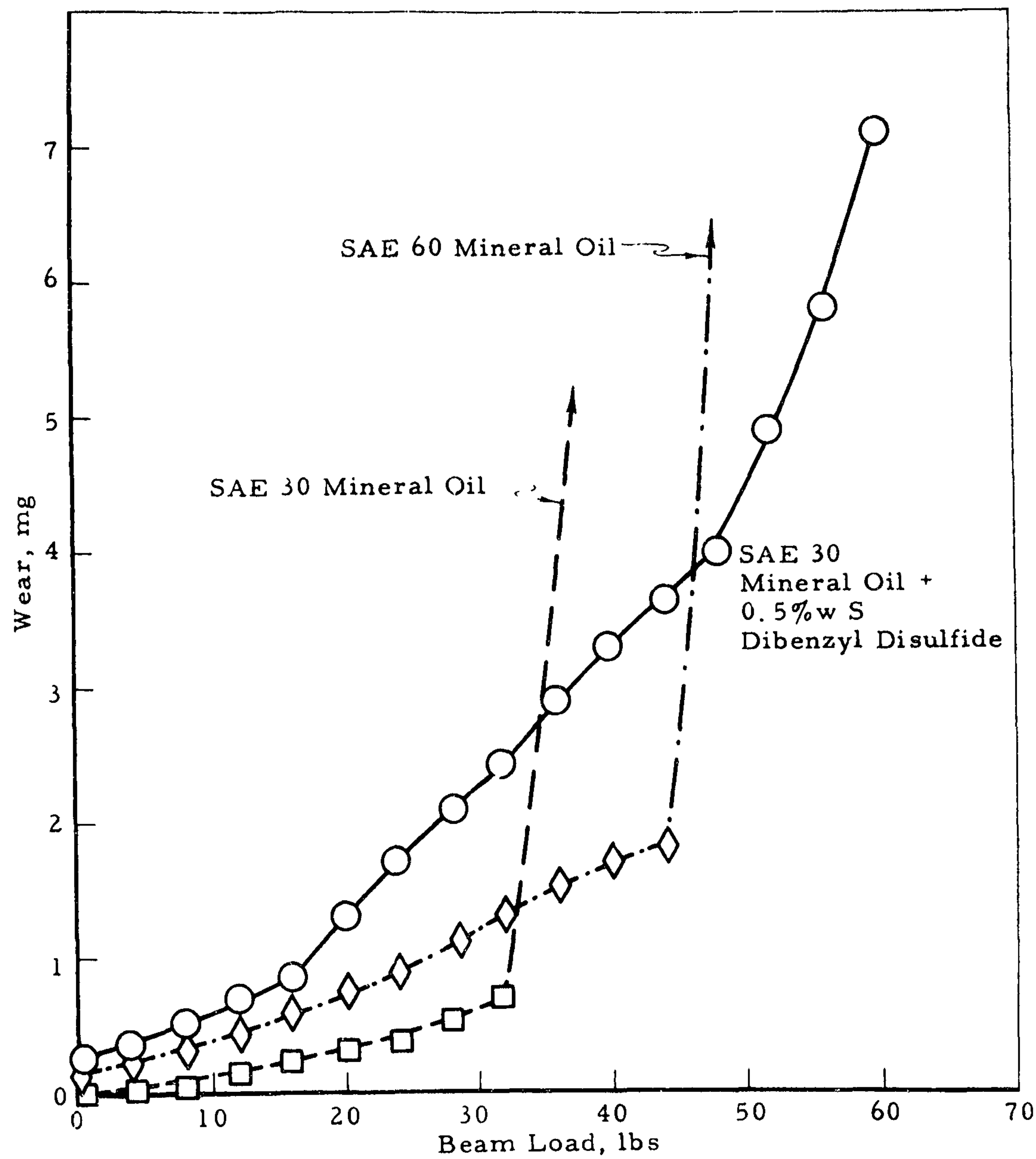

Figure 37. WEAR AT 1070 RPM

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Scoring Performance of GTO-38 and Wright Air Development Center Reference Oil "B"

On the request of Wright Air Development Center the scoring performances of several oils meeting the MIL-L-25336 specification have been investigated. The oils were GTO-38 and Wright Air Development Center reference oil "B". The load carrying capacity of these oils were determined at speeds of 3200, 5000, 10,000, 15,000 and 20,000 rpm. The tests were performed in the Shell Development High Speed Spur Gear machine. Supply test oil temperature was 100°F, the flow-rate was 10 cc/sec, and the load increments were 4 lb beam load. The duration of running at each load setting was 5 minutes. The gears used were our standard 17-19 tooth test gears of 6 diametral pitch, 20° pressure angle and 0.25 inch face width. These gears were of the batches received in 1956 which were of somewhat poorer tolerances than specified (see page 2, bimonthly progress report No. 20). To insure better accuracy the gears for these tests were hand-picked; only the gears with eccentricity and spacing errors less than 0.0004 inch were used. This limited the number of test gears available and only two tests were made at each operating conditions, instead of eight as previously was required by the Wright Air Development Center.

The results of these tests are presented in Table 15 and are plotted in Figure 38 and Figure 39.

Discussion of Results. The score loads in Table 15 are presented in terms of beam load and as load per inch of face of the gears. It is to be noted that the reproducibility of these tests was within 4 lb beam load, except for the tests with GTO-38 oil at 3200 rpm where reproducibility was 8 lb.

Both oils showed a higher load carrying capacity compared with previously tested uncompounded mineral and synthetic oils of similar viscosity. Also, the shapes of the load carrying capacity - speed curves differ from those of uncompounded oils. It appears, therefore, that both of these oils, GTO-38 and WADC reference oil "B", are fortified with an extreme pressure agent.

Table 16. SCORING PERFORMANCE OF GTO-38 AND WADC REFERENCE OIL "B"

Lubricants	3200 rpm		5000 rpm		10,000 rpm		15,000 rpm		20,000 rpm	
	Beam lb	lb/in	Beam lb	lb/in	Beam lb	lb/in	Beam lb	lb/in	Beam lb	lb/in
GTO-38	44	3,120	40	2,830	56	3,960	52	3,680	48	3,400
	52	3,680	44	3,120	52	3,680	48	3,400	48	3,400
WADC Reference Oil "B"	32	2,270	32	2,270	44	3,120	36	2,550	44	3,120
	28	1,980	36	2,550	44	3,120	36	2,550	48	3,400
	32	2,270								
	32	2,270								

Figures 38 and 39 follow

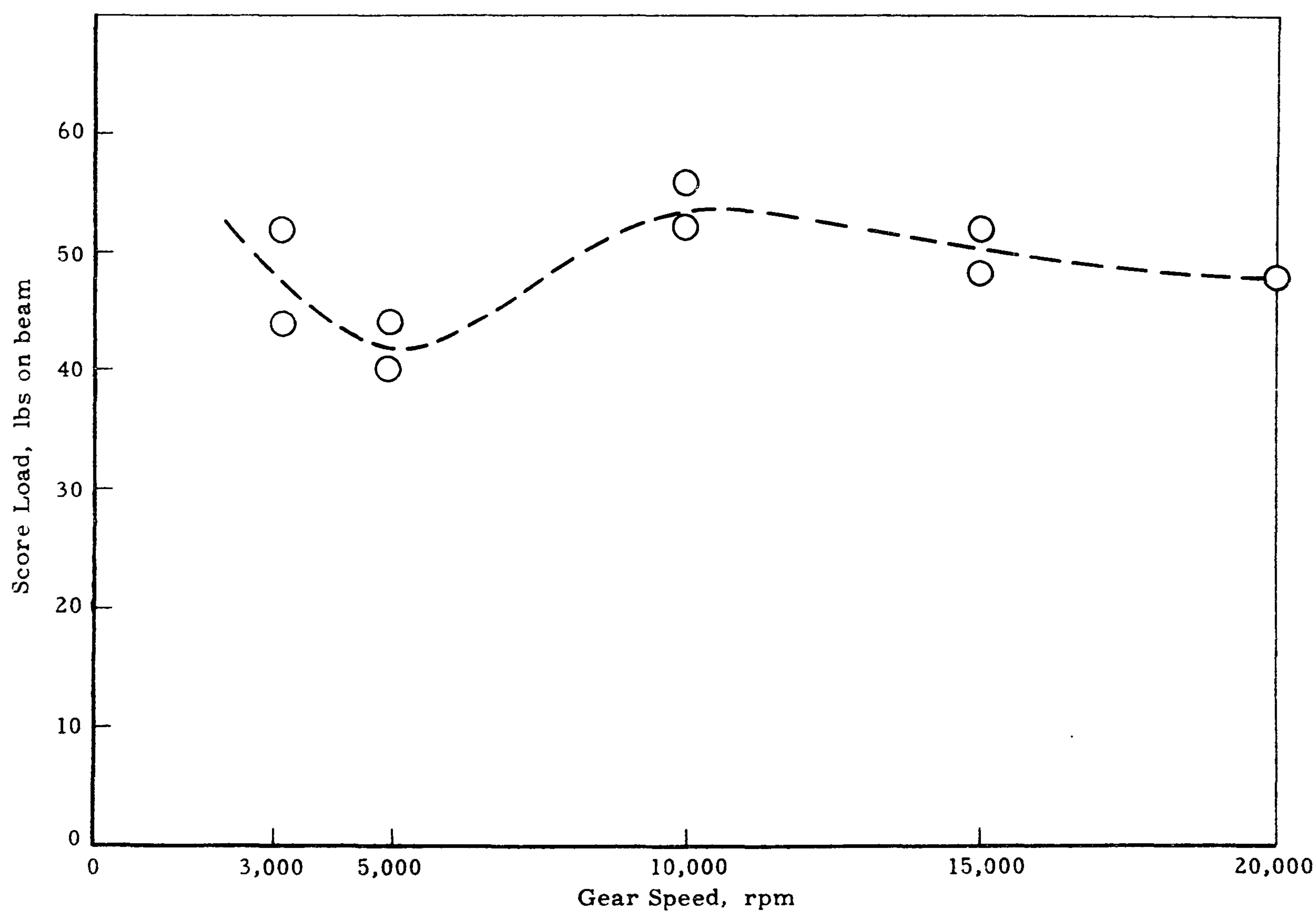

Figure 38. SCORING PERFORMANCE OF GTO-38 OIL AT VARIOUS SPEEDS

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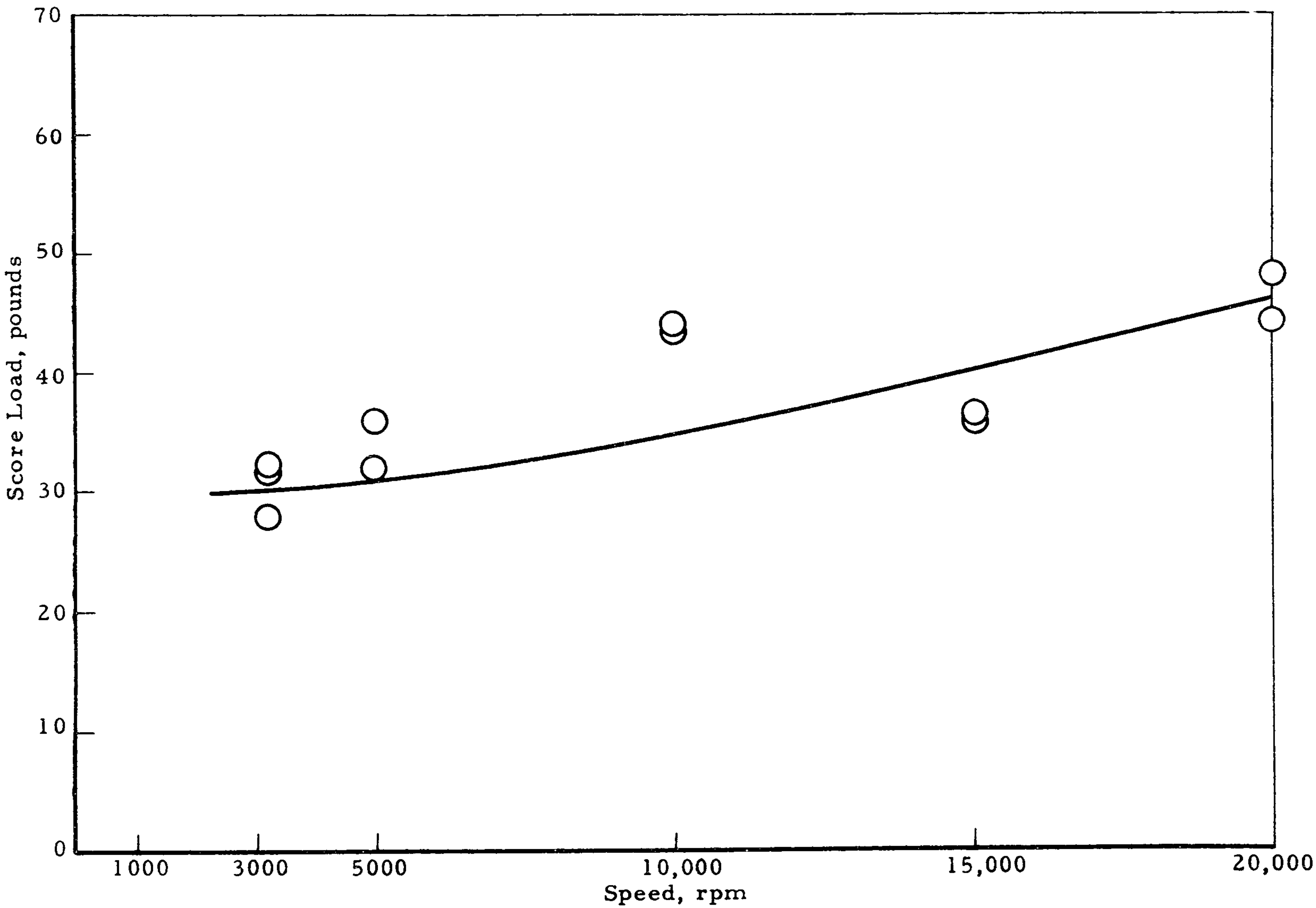

Figure 39. SCORING PERFORMANCE OF WADC REFERENCE OIL "B"-LOAD CARRYING CAPACITY VS SPEED

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Studies of Extreme Pressure Lubrication

Formation and Attrition of an E.P. Film

Test Lubricant and Procedure. During the first year of our experiments with tagged extreme pressure compounding agents highly informative results were obtained. The existence of the extreme pressure films on the gear teeth working surfaces was confirmed and the distribution and thickness of these films were determined. It was learned also that the thickness of such films increased with the increase in load and that for each set of operating conditions an appropriate steady state thickness was extablished. From these results it was postulated that extreme pressure films are being formed and attrited continuously while the gears are in operation. The following year was devoted to the studies of the mechanism of attrition of extreme pressure films. These experiments were also fruitful. Thus, it was learned that extreme pressure films previously established on gear teeth became gradually attrited when the gears were operated with an unreactive oil and that this attrition is fast under a sliding motion but is unaffected by pure rolling. The autoradiographs of the gear tooth working surfaces also suggested that the extreme pressure films are displaced in the direction of sliding, thus exibiting plastic characteristics; but these autoradiographs were weak and often of insufficient clearness to permit a more positive conclusion. Obviously, to obtain a sharper replica of the teeth working surfaces the radioactivity of the test extreme pressure lubricant should be increased. This was done in the series of experiments performed during the period covered by this report.

Most of our previous studies of formation and attrition of extreme pressure films were performed with mineral oils compounded with dibenzyl disulfide tagged with the radioactive isotope of sulfur, S^{35}. The radioactivities of these oils were usually adjusted to give approximately 30 cpm/cm^2/monolayer of the resultant film, when counting was performed under the standardized geometry. The new sample of oil for the present series of experiment was adjusted to 82 cpm/cm^2/monolayer or approximately 2.7 times higher radioactivity than of the previously used batches.

The experiments were performed in the Slow Speed Spur Gear Machine using fresh test gears in one series of tests and reground gears in another. An extreme pressure film was built-up on the working surfaces of the gear teeth by operating the gears first with the tagged extreme pressure lubricant. In these runs the load was increased by 5 lb steps, running one hour at each load setting. After attaining a load of 25 lb and repeating the run at 15 lb load for two hours, the lubricant was changed to uncompounded SAE 30 mineral oil, and the same gears were operated at 15 lb beam load, varying the duration of running from five minutes to six hours. After each run, the extent and the distribution of the radioactivity on the working surfaces were studied using both counting and autoradiography techniques.

The results of these experiments are presented in Tables 17 to 19 and graphically in Figures 40 to 52. In Figures 42 to 45 examples of the autoradiographs are given. Figures 46 to 49 contain typical traces of the optical densities of the autoradiographs of the working surfaces.

Discussion of Results. The data given in Table 17 and Table 18 and plotted in Figure 40 and Figure 41 represent the results obtained by direct counting with a thin window GM counter. The geometry of counting was similar to that shown in Figure 27 of the Summary Report S-13694. Due to the higher radioactivity of the lubricant the counts were higher than in the previous experiments. The results obtained, however, were in full agreement with those of the previously performed tests. Again the counts, which should be proportional to the film thickness, increased with load and were somewhat higher on the 17 tooth gear than on the 19 tooth gear. In the attrition runs, the radioactivity emitted by the teeth working surfaces decreased with time of running, indicating that the E.P. film was gradually attrited. The rate of this attrition was high at the beginning and decreased with time. Approximately 1/3 of the E.P. film remained on the working surface. The results obtained with the counting technique were alike for both types of gears, the fresh one and the reground one.

The gears were autoradiographed after running at each load with the extreme pressure lubricant and after each period of time during the attrition part of these experiments. These autoradiographs are presented in Figures 42 to 45. The autoradiographs are clearer and sharper than previously obtained. The optical densities of these autoradiographs were measured with the Spinco Model R, Analytrol described on page 29 of the Summary Report S-13694. The summary of these tracings are presented in Figures 46 to 49. It is of interest that the fresh test gears gave tracings with well defined addendum, pitch circle and dedendum, while tracings of the reground gears are irregular and jagged, suggesting that the surfaces of these gears were of a poorer finish. On account of these irregularities, the data obtained by the autoradiography technique from the fresh gears only were interpreted and used. These data are presented in Table 19 and are plotted in Figures 50 and 51. The optical densities of the autoradiographs at the addendum, the pitch circle and the dedendum were established and converted into radioactivity by means of the chart given in Figure 52. The radioactivity, expressed in cpm/cm^2, was then converted into the film thickness expressed as a number of layers. The plots of these data show that for both the 17 tooth and the 19 tooth gears the thinnest film occurred on the pitch circle, where the relative motion is pure rolling. These plots also show that thicknesses of the extreme pressure films increased with the increase in load and that during the attrition runs the thickness of these films decreased at a constantly decreasing rate. The graphs presented in Figure 47 suggest that during the attrition runs not only did the film thickness decrease, but also the locations of the maximum points were gradually displaced in the direction of sliding. The data also shows that approximately 1/3 of these films remained intact. Comparison of these data with those obtained by direct counting shows that both techniques gave similar results. The advantages of the autoradiography are that it supplies additional information on the distribution and thickness of extreme pressure films. Here it should be pointed out that our calculations of film thickness are based on an assumption that the films are uniformly distributed over the localized areas and that the rise of the radioactivity is caused by the growth of the films in depth. However, the possibilities that a portion of the rise of radioactivity with increase in load could be caused by an increase of actual contacting area are not excluded. Unfortunately no methods are known for the elucidation of the problem of the true nature of contact. In connection with this, the results of attrition experiments showing that at a load of 15 lb 1/3 of the radioactivity remained intact are of interest. If one visualizes a surface as consisting of hills and valleys and assumes that the true contact area is proportional to the load, then one could speculate that in the case of our gears operating at 15 lb beam load the true contact area is approximately 2/3 of the total area. This conclusion is made on an assumption that the remaining radioactivity is emmitted by the extreme pressure films formed on the floors of the valleys and therefore not subjected to direct attrition by the mating surface. It is realized, however, that much additional evidence is necessary before the above speculation can be confirmed or discarded.

Conclusions. 1. The results of these experiments confirm the findings of our previous tests as follow: (a) E.P. film thickness increases with an increase in load. (b) The E.P. film is thinnest at the pitch circle and its thickness increased toward the addendum and the dedendum. (c) When gears with an established E.P. film are operated with uncompounded oil, the E.P. film becomes gradually attrited. The rate of this attrition decreases with time. (d) At 15 lb beam load, approximately 1/3 of the E.P. film remains intact. It is postulated that this portion of the film is located on the floors of hypothetical valleys. (e) Extreme pressure films are not only attrited but are also displaced in the direction of sliding.

2. The uneven jagged appearance of the optical density curves of the autoradiographs of reground gears suggests a poorer surface finish for these gears.

Table 17. FORMATION AND ATTRITION OF AN E.P. FILM DIRECT COUNTING WITH GEIGER COUNTER

Fresh Test Gears

Lubricant	Load, lb	Time, min	BG cpm	17 Tooth Gear					19 Tooth Gear				
				Readings, cpm			Avg cpm	Corrected for BG	Readings, cpm			Avg cpm	Corrected for BG
				1	2	3			1	2	3		
SAE 30 Containing DBDS[35]	0	60	28	45	42	36	41	13	70	62	63	65	37
	5	60	21	105	109	98	104	83	131	116	122	123	102
	10	60	20	164	131	134	143	123	181	159	154	165	145
	15	60	23	352	315	340	336	313	305	286	272	288	265
	20	60	22	738	650	705	698	676	578	507	505	530	508
	25	60	27	835	865	817	839	812	659	633	680	657	630
	15	120	21	894	951	844	896	875	582	619	536	579	558
SAE 30	15	5	29	704	791	823	775	744	493	533	592	539	510
	15	10	47	730	721	706	719	672	431	489	642	521	474
	15	10	26	592	608	614	670	644	353	457	500	437	411
	15	20	24	593	571	564	576	552	309	362	414	362	338
	15	30	26	490	464	509	488	462	340	410	395	382	356
	15	60	41	456	440	449	448	407	281	391	318	330	289
	15	360	26	370	369	366	368	342	222	261	280	254	228

Table 18. FORMATION AND ATTRITION OF AN E. P. FILM DIRECT COUNTING WITH GEIGER COUNTER

Reground Test Gears

Lubricant	Load, lb	Time, minutes		BG cpm	17 Tooth Gear					19 Tooth Gear				
		During Run	Total		Readings, cpm			Avg cpm	Corrected for BG	Readings, cpm			Avg cpm	Corrected for BG
					1	2	3			1	2	3		
SAE 30 Containing DBDS^{35}	0	60	60	20	33	36	37	35	15	34	30	35	33	13
	5	60	120	23	77	70	72	73	50	61	61	63	62	39
	10	60	180	25	137	151	151	146	121	108	107	113	109	84
	15	60	240	26	397	383	370	383	357	194	186	213	198	172
	20	60	300	22	743	768	781	764	742	346	396	387	376	354
	25	60	360	28	911	933	925	723	695	477	502	483	487	459
	15	120	480	25	943	950	849	914	889	416	413	393	411	386
SAE 30	15	5	5	28	820	879	822	840	812	352	347	400	366	338
	15	10	15	54	757	852	721	743	689	340	376	355	357	303
	15	10	25	25	657	650	594	634	609	259	290	314	288	263
	15	20	45	25	561	609	545	572	547	261	279	300	280	255
	15	30	75	44	500	575	546	540	496	236	224	276	245	201
	15	60	135	24	476	512	475	488	464	219	221	257	232	208
	15	360	495	26	398	439	379	405	379	162	190	187	180	154

Table 19. FORMATION AND ATTRITION OF AN E. P. FILM BY AUTORADIOGRAPH METHOD

Fresh Test Gears

Lubricant	Load lb	Time min	Photo No.	17 Tooth Gear Film No. 1									19 Tooth Gear Film No. 2								
				Addendum			Pitch Circle			Dedendum			Addendum			Pitch Circle			Dedendum		
				OD	cpm/cm²	No. of Layers	OD	cpm/cm²	No. of Layers	OD	cpm/cm²	No. of Layers	OD	cpm/cm²	No. of Layers	OD	cpm/cm²	No. of Layers	OD	cpm/cm²	No. of Layers
SAE 30 Containing DBDS[35]	0	60	1	0.3	42	0.51	0.2	27	.24	0.5	91	1.11	0.6	118	1.43	0.3	42	0.51	1.6	545	6.45
	5	60	2	0.65	135	1.64	0.35	55	.67	1.05	280	3.4	1.2	342	4.16	1.1	300	3.66	2.4	1050	12.8
	10	60	3	0.90	220	2.68	0.5	91	1.11	1.40	435	5.3	1.7	600	7.3	1.3	390	4.76	2.0	790	9.64
	15	60	4	2.00	790	9.64	1.40	435	5.3	2.65	1240	13.2	3.1	1570	19.2	1.6	545	6.65	1.6	545	6.45
	20	60	5	2.85	1370	16.7	2.30	980	11.9	3.05	1540	18.8	3.6	2000	24.4	2.6	1190	14.5	3.7	2120	25.8
	25	60	6	4.00	2400	29.2	3.00	1490	18.2	3.50	1920	23.4	5.4	3950	48.2	1.9	725	8.85	3.0	1490	18.2
	15	120	7	3.65	2050	25.0	2.70	1270	15.5	3.10	1570	19.1	4.1	2500	30.5	1.2	345	4.21	5.1	3600	43.9
SAE 30	15	5	8	3.30	1740	21.2	2.70	1270	15.5	2.90	1430	17.4	4.2	2600	31.7	0.8	185	2.25	5.0	3450	42.1
	15	10	9	2.90	1430	17.4	2.40	1050	12.8	2.70	1270	15.5	3.0	1490	18.2	0.9	220	2.68	2.5	1120	13.65
	15	10	10	2.80	1340	16.3	2.20	910	11.2	2.60	1190	14.5	3.2	1650	20.05	0.6	118	1.43	4.5	2900	35.4
	15	20	11	2.80	1340	16.3	2.20	910	11.2	2.60	1190	14.5	2.2	910	11.2	0.8	185	2.25	2.3	980	11.95
	15	30	12	2.40	1050	12.8	1.90	730	8.9	2.20	910	11.2	2.2	910	11.2	0.5	91	1.11	-	-	-
	15	60	13	2.20	910	11.2	1.70	600	7.3	2.00	790	9.6	2.50	1120	13.7	0.4	65	0.79	2.2	910	11.1
	15	360	14	1.80	660	8.05	1.30	390	4.76	1.60	545	6.65	2.10	850	10.4	0.3	42	0.51	1.7	605	7.85

Figures 40 through 52 follow

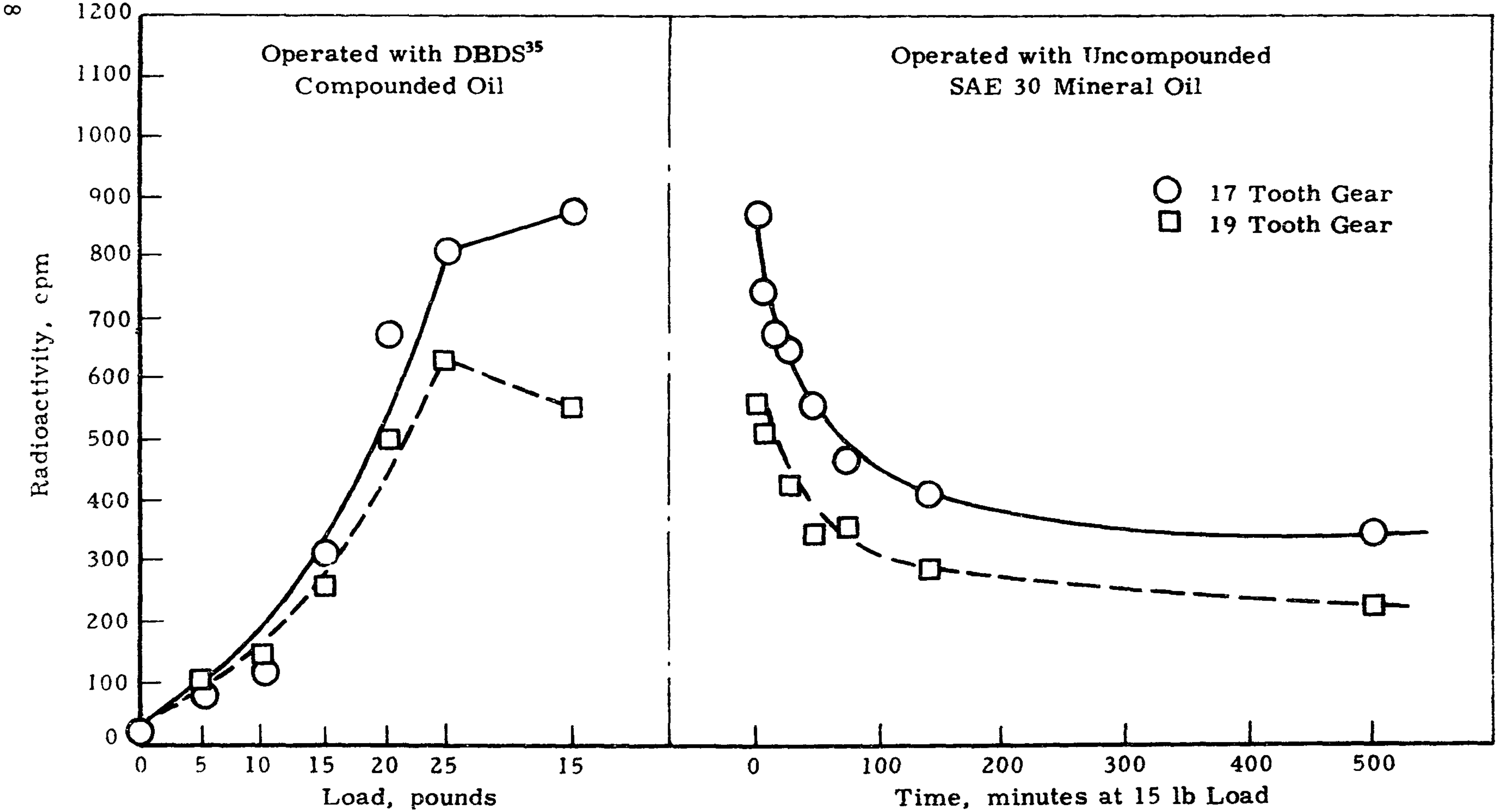

Figure 40. FORMATION AND ATTRITION OF AN EXTREME PRESSURE FILM
Gears - Fresh Test Gears
Radioactivity Measured by Geiger Counter

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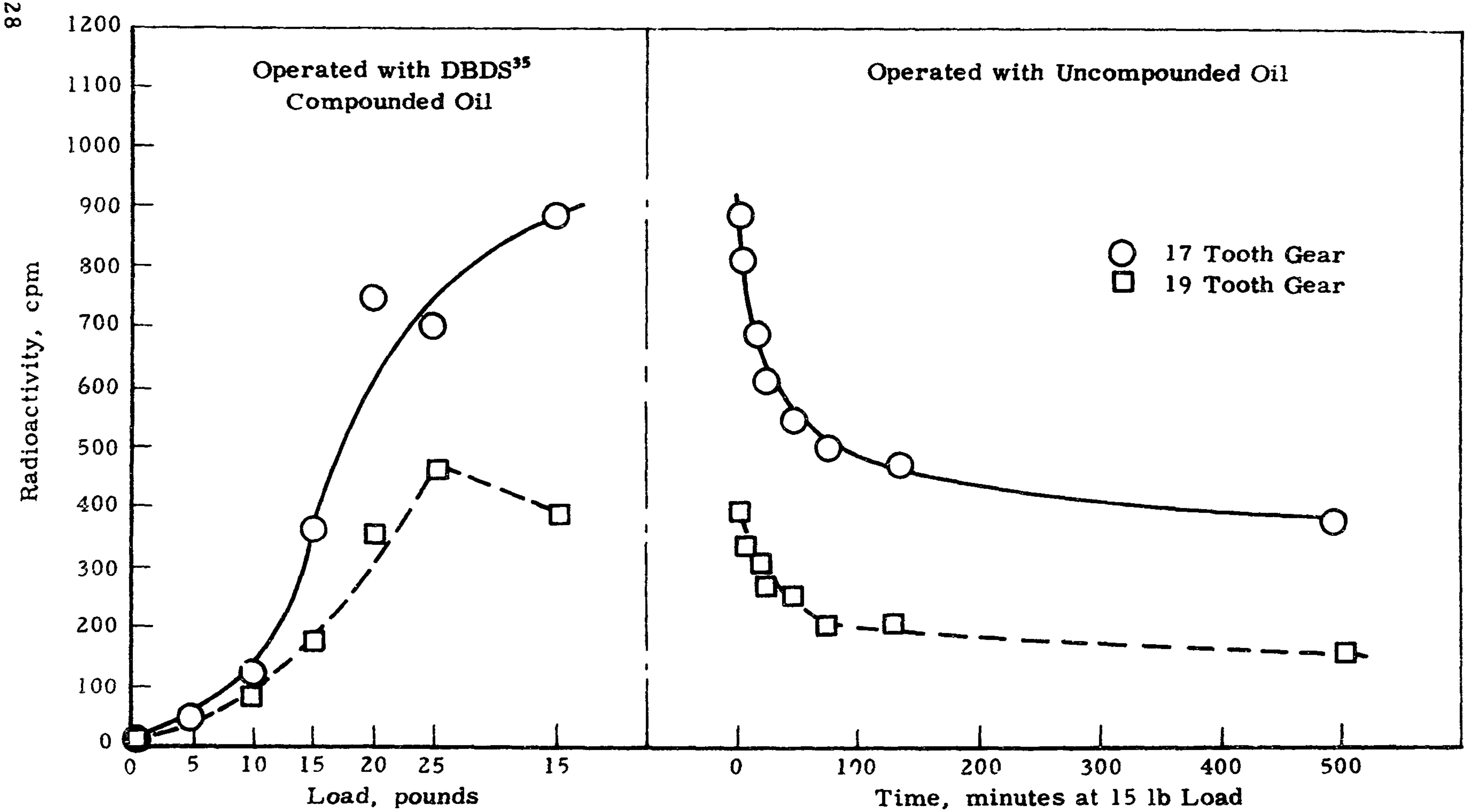

Figure 41. FORMATION AND ATTRITION OF AN EXTREME PRESSURE FILM
Gears - Reground Test Gears

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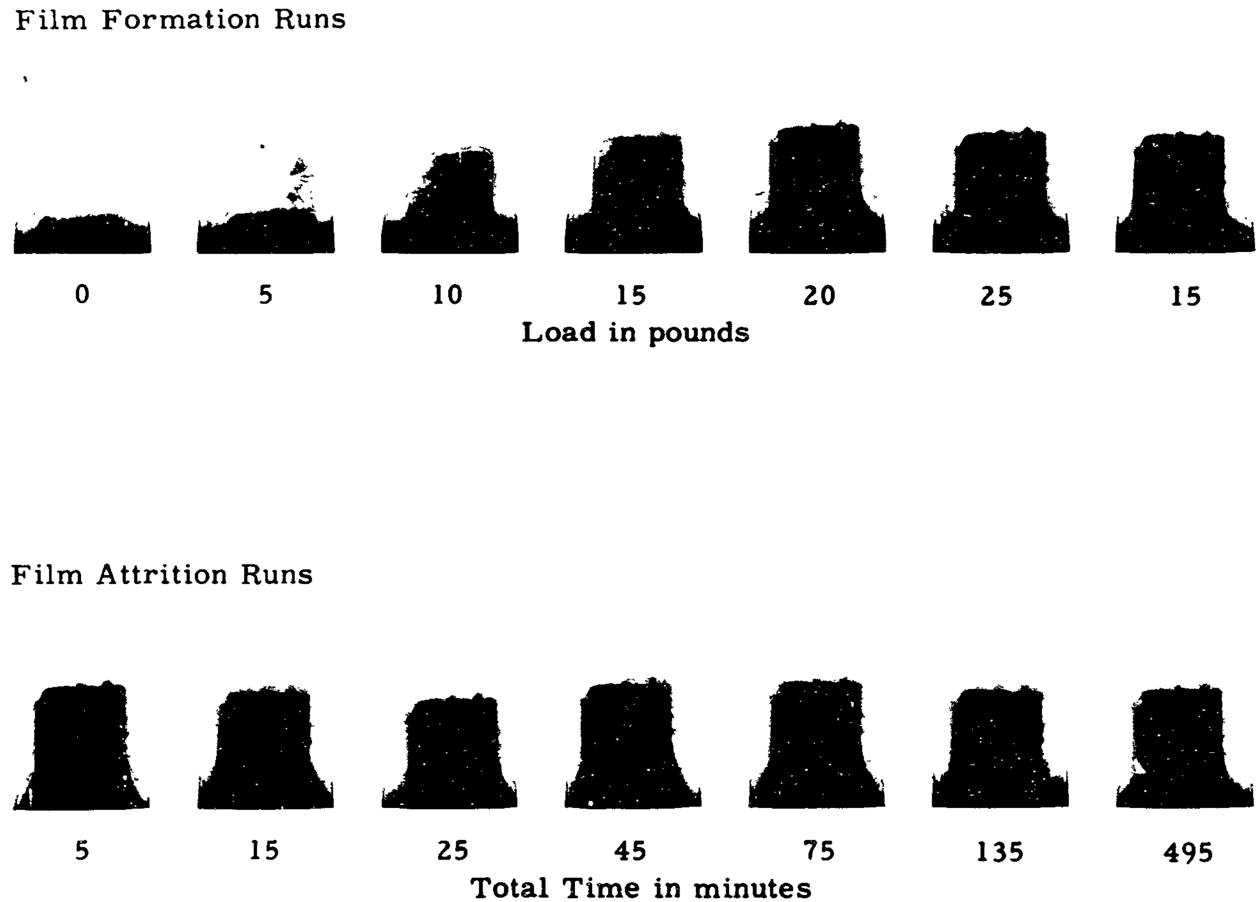

Figure 42. AUTORADIOGRAPHS OF 17 TOOTH FRESH GEAR

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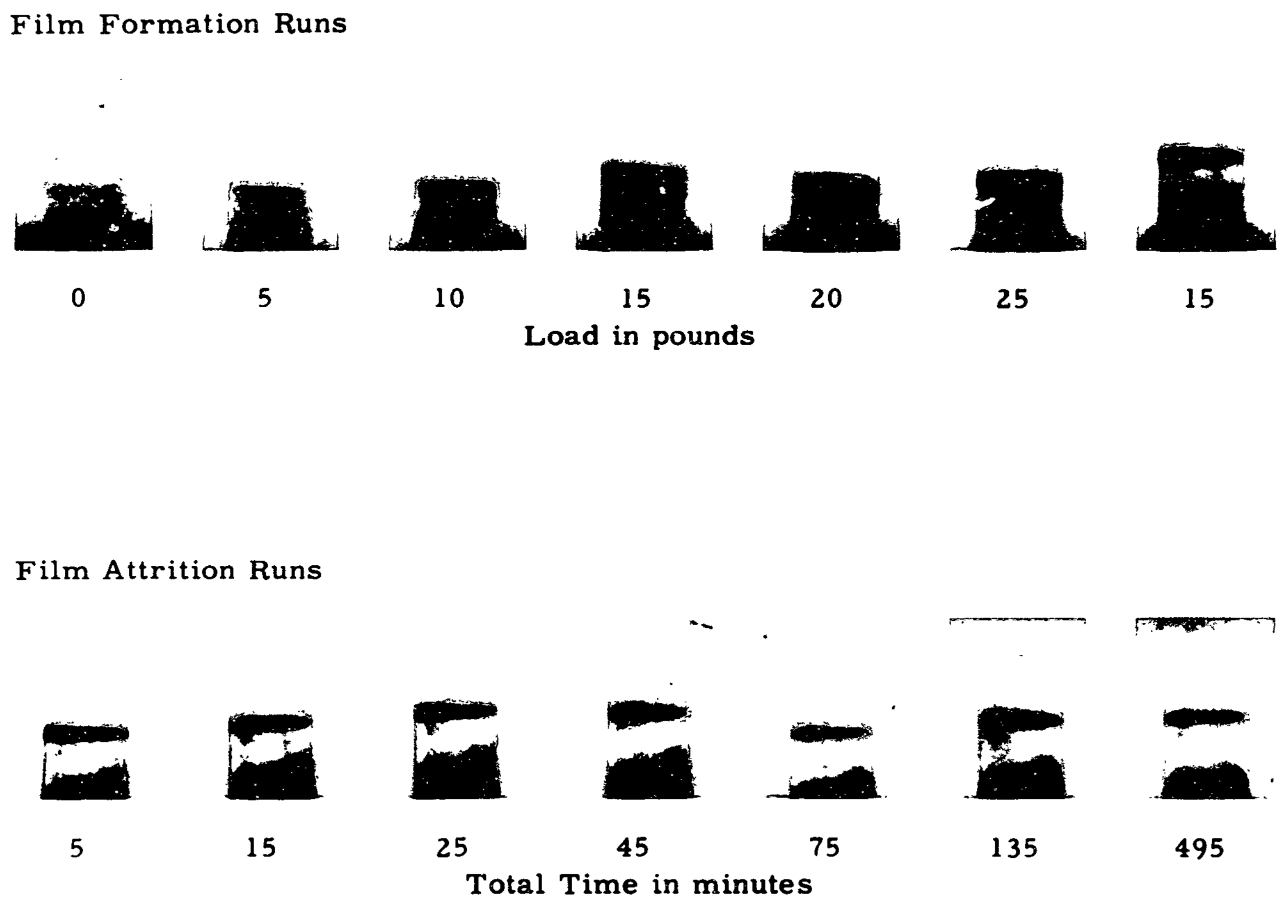

Figure 43. AUTORADIOGRAPHS OF 19 TOOTH FRESH GEAR

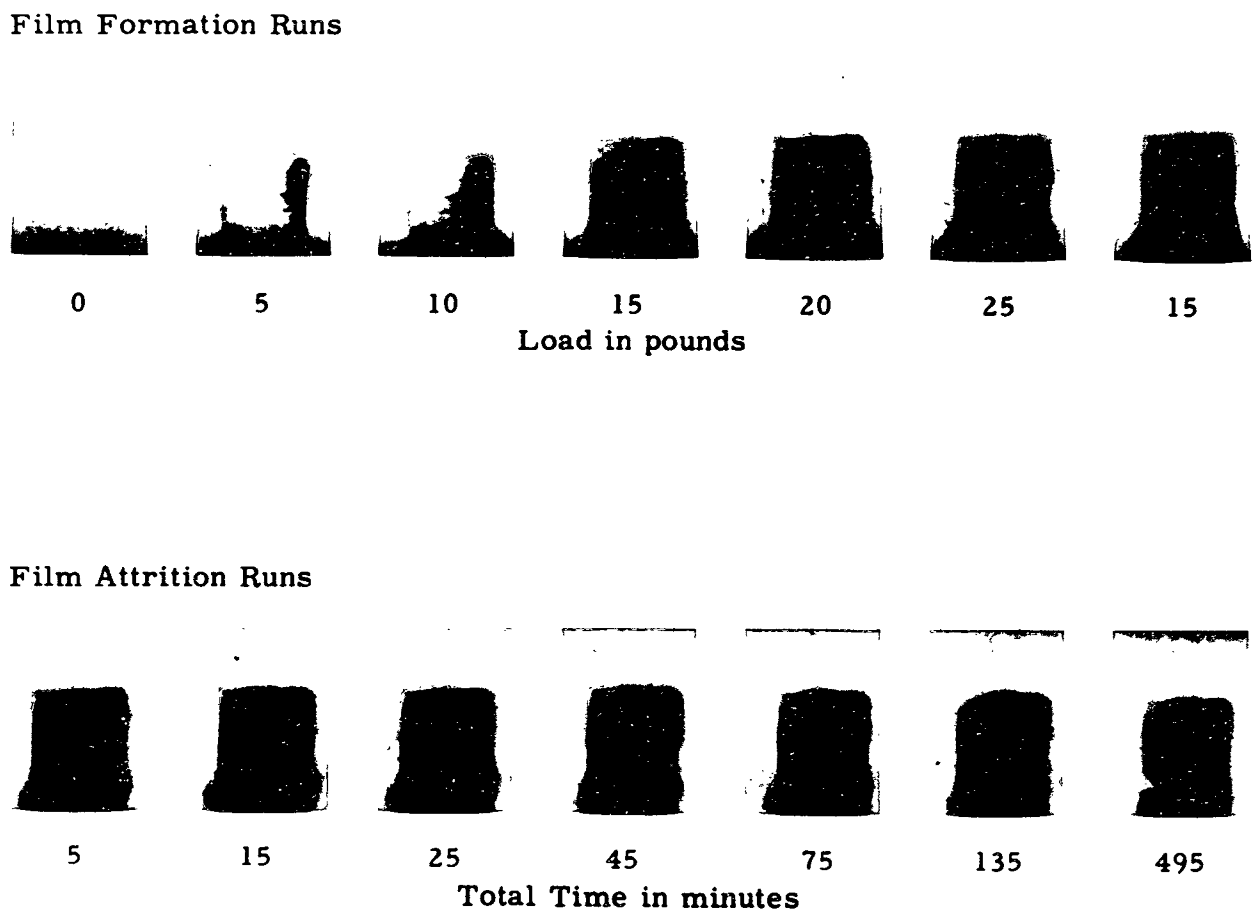

Figure 44. AUTORADIOGRAPHS OF 17 TOOTH REGROUND GEAR

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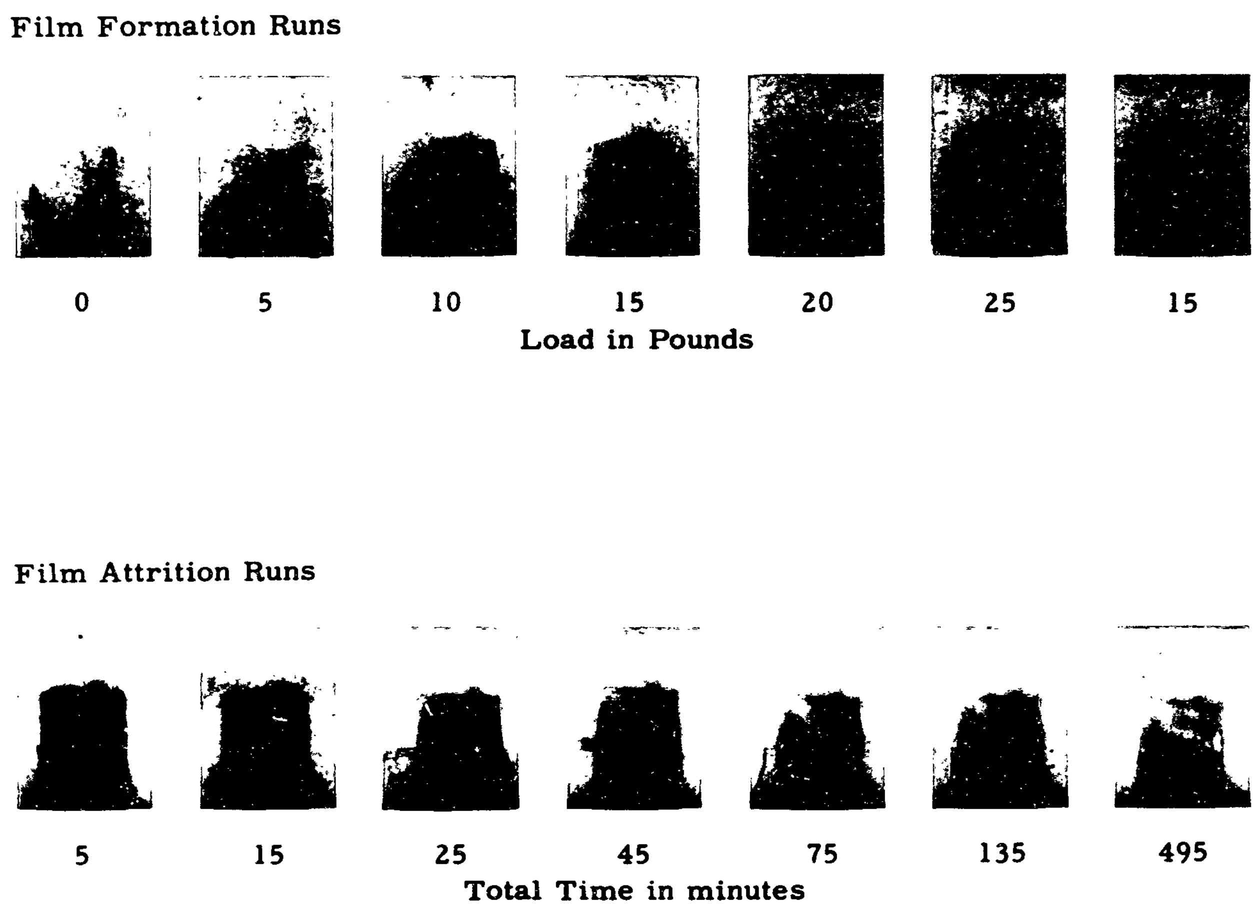

Figure 45. AUTORADIOGRAPHS OF 19 TOOTH REGROUND GEAR

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Figures 46, 47, 48 and 49

ORIGINAL CONTAINS COLOR PLATES: ALL ASTIA REPRODUCTIONS WILL BE IN BLACK AND WHITE.

1-1 0 lb Beam Load
1-2 5 lb Beam Load
1-3 10 lb Beam Load
1-4 15 lb Beam Load
1-5 20 lb Beam Load
1-6 25 lb Beam Load

1-7 15 lb Beam Load
1-8 5 min Attrition at 15 lb Beam Load
1-9 15 min Attrition at 15 lb Beam Load
1-10 25 min Attrition at 15 lb Beam Load
1-11 45 min Attrition at 15 lb Beam Load
1-12 75 min Attrition at 15 lb Beam Load
1-13 135 min Attrition at 15 lb Beam Load
1-14 495 min Attrition at 15 lb Beam Load

17 Tooth Pinion
MB Gear

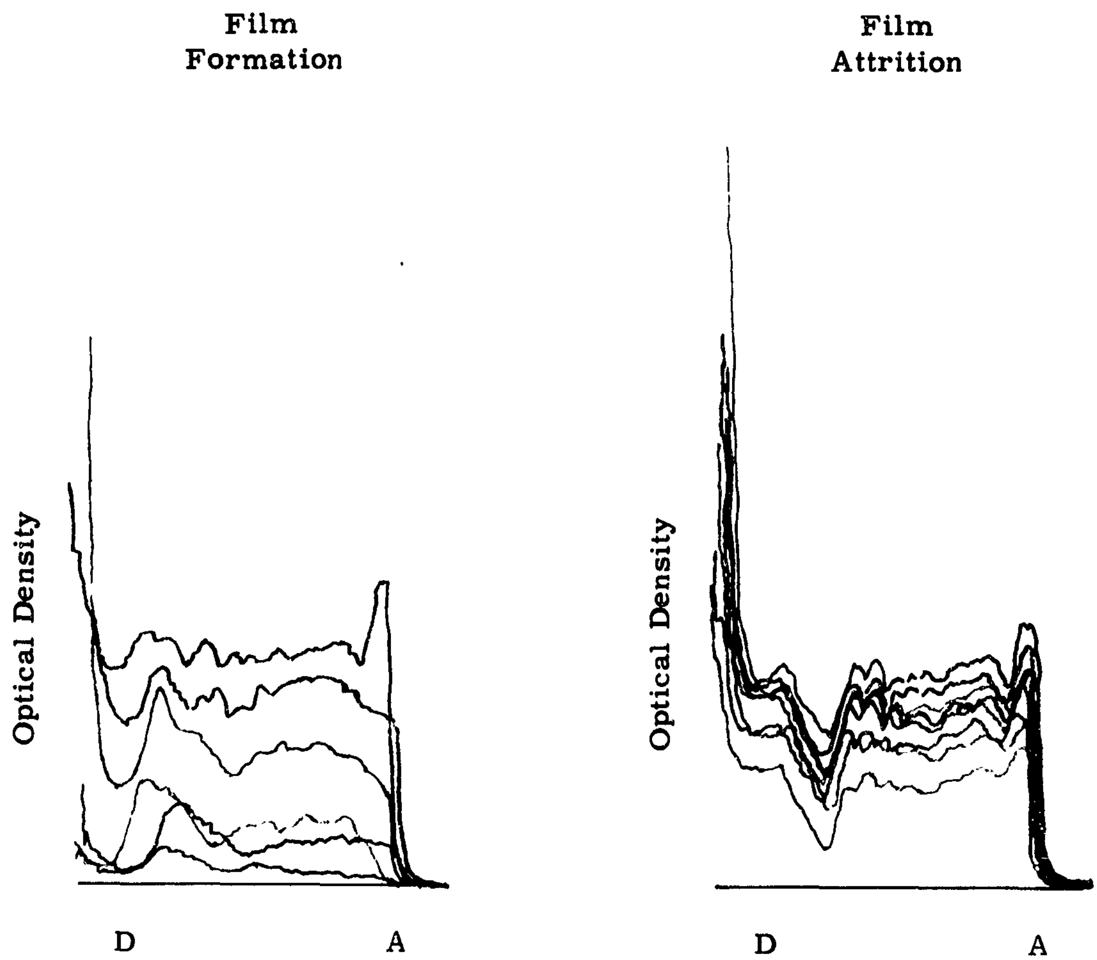

Figure 46.

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2-1 0 lb Beam Load
2-2 5 lb Beam Load
2-3 10 lb Beam Load
2-4 15 lb Beam Load
2-5 20 lb Beam Load
2-6 25 lb Beam Load

2-7 15 lb Beam Load
2-8 5 min Attrition at 15 lb Beam Load
2-9 15 min Attrition at 15 lb Beam Load
2-10 25 min Attrition at 15 lb Beam Load
2-11 45 min Attrition at 15 lb Beam Load
2-12 75 min Attrition at 15 lb Beam Load
2-13 135 min Attrition at 15 lb Beam Load
2-14 495 min Attrition at 15 lb Beam Load

19 Tooth Gear
MB Gear

Film Formation

Film Attrition

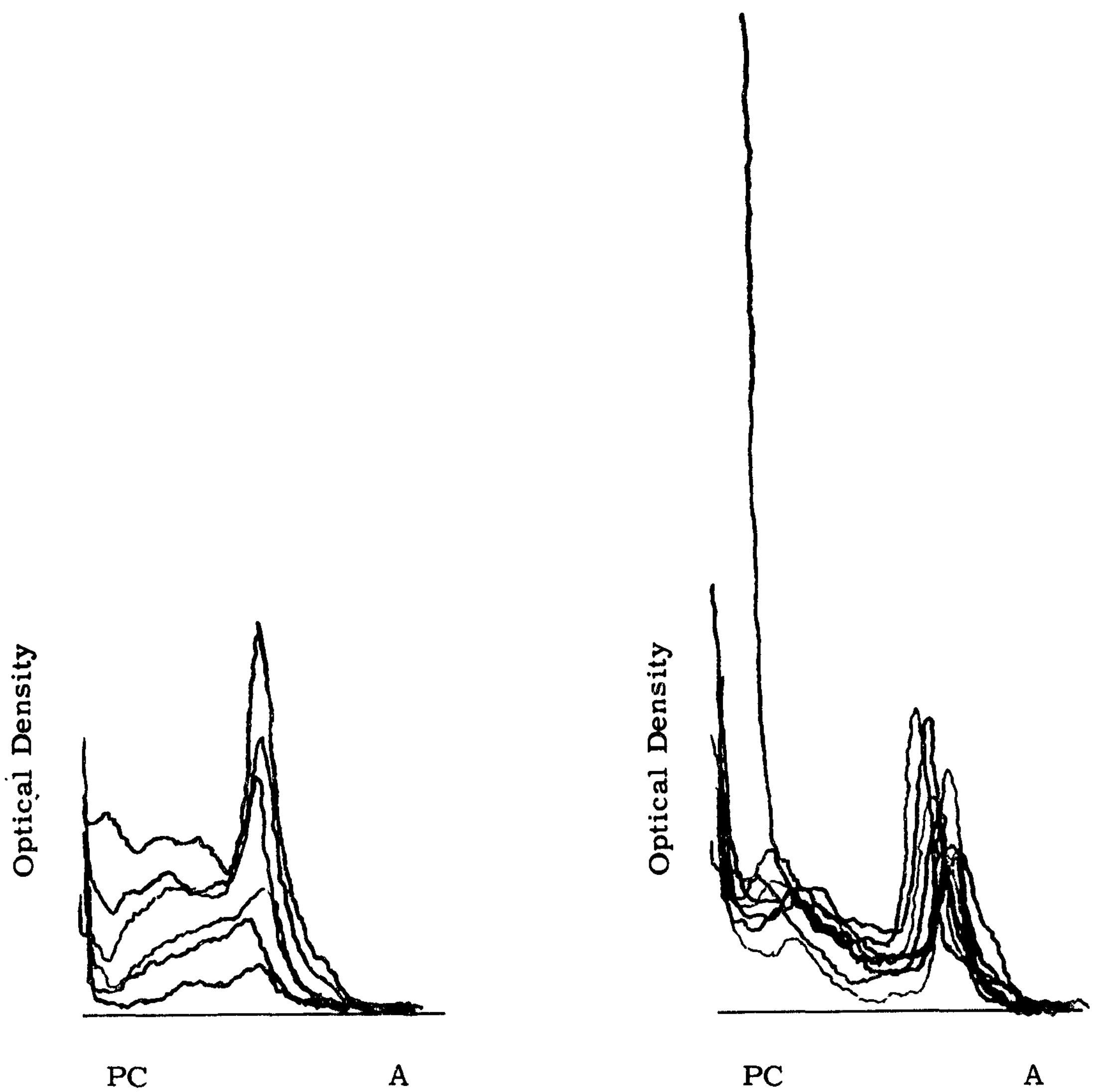

Figure 47.

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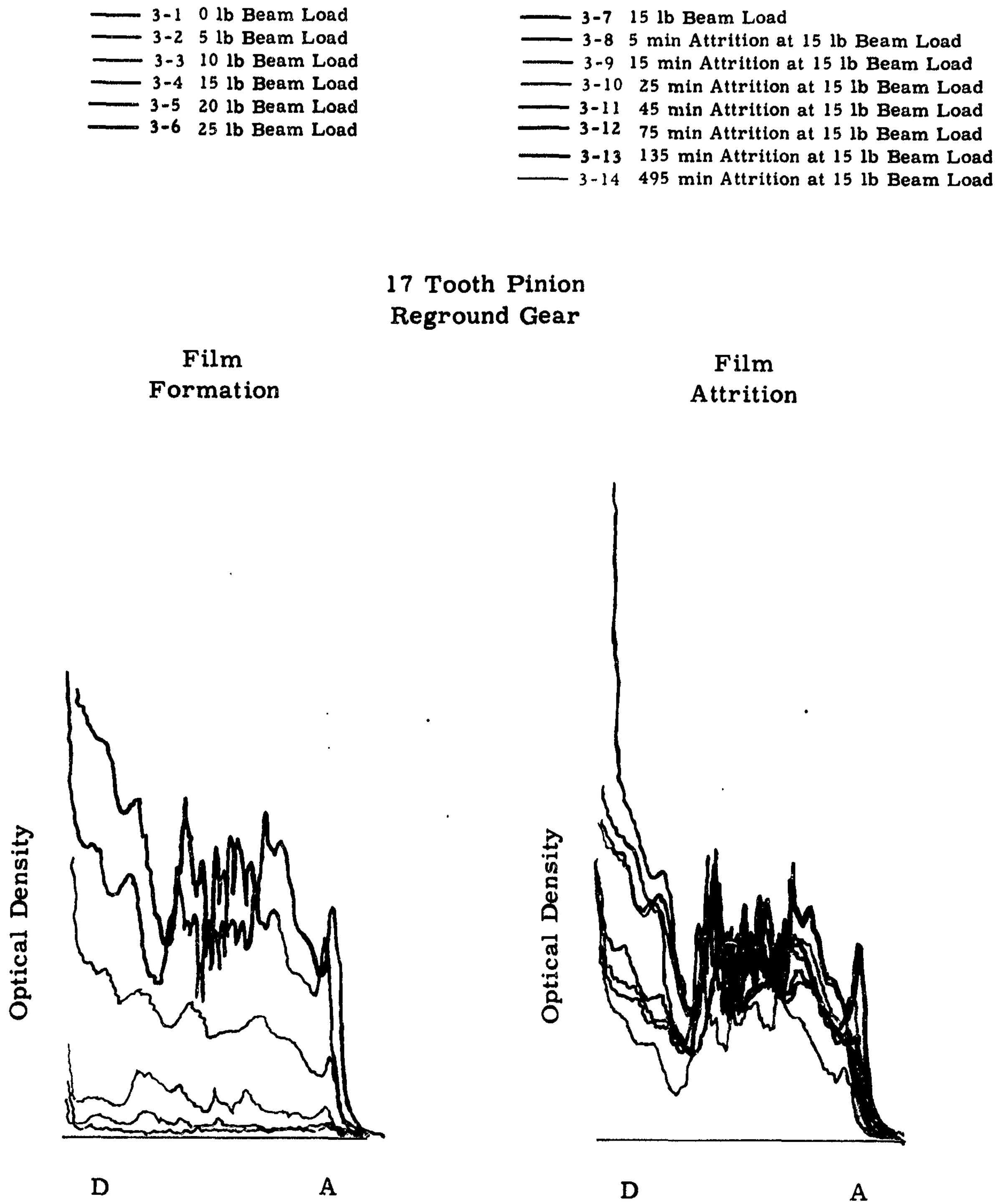
3-1 0 lb Beam Load
3-2 5 lb Beam Load
3-3 10 lb Beam Load
3-4 15 lb Beam Load
3-5 20 lb Beam Load
3-6 25 lb Beam Load
3-7 15 lb Beam Load
3-8 5 min Attrition at 15 lb Beam Load
3-9 15 min Attrition at 15 lb Beam Load
3-10 25 min Attrition at 15 lb Beam Load
3-11 45 min Attrition at 15 lb Beam Load
3-12 75 min Attrition at 15 lb Beam Load
3-13 135 min Attrition at 15 lb Beam Load
3-14 495 min Attrition at 15 lb Beam Load
17 Tooth Pinion
Reground Gear
Film
Formation
Film
Attrition
Optical Density
Optical Density
D
A
D
A

Figure 48.

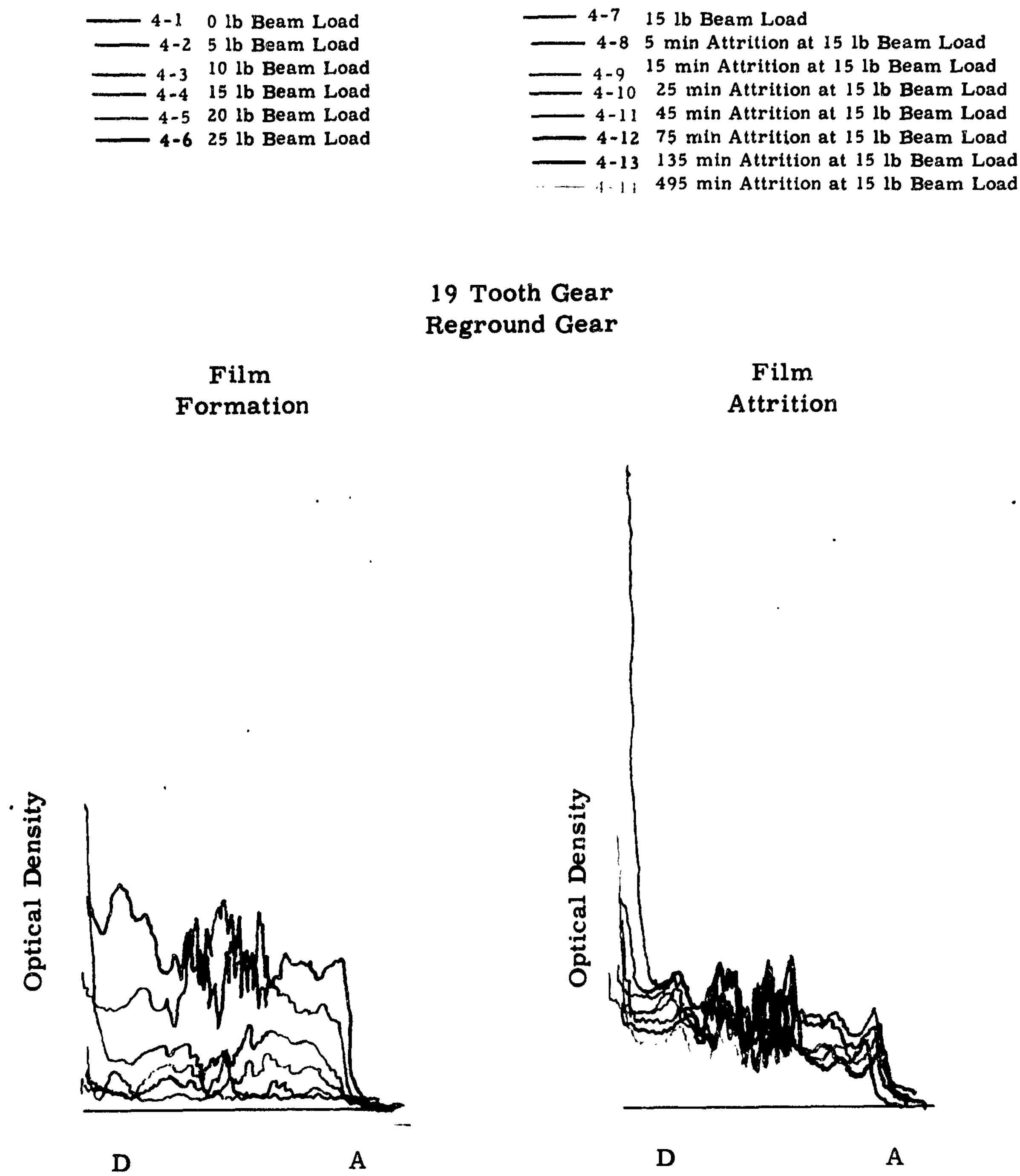

Figure 49.

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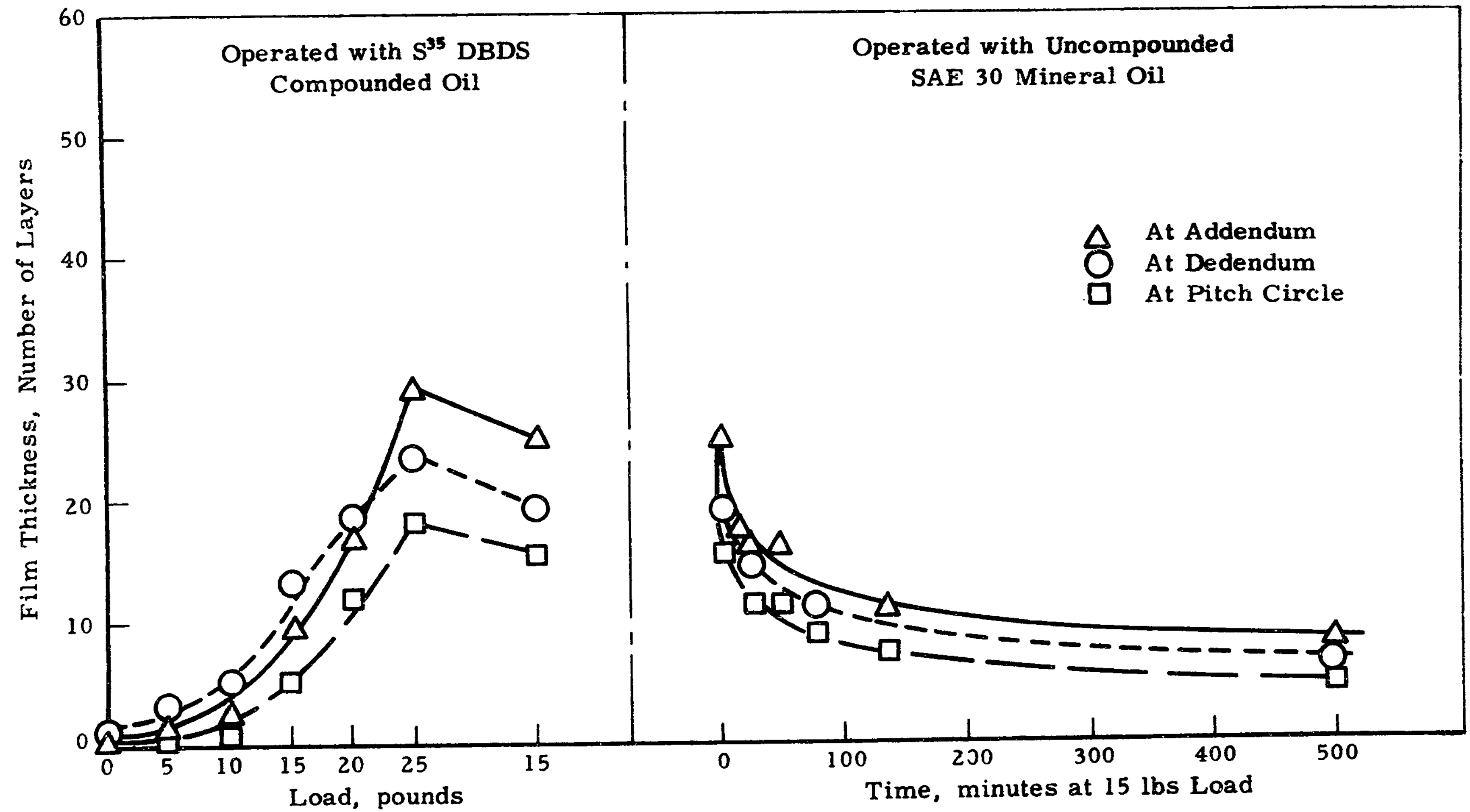

Figure 50. FORMATION AND ATTRITION OF AN EXTREME PRESSURE FILM

Gear - 17 Tooth Test Gear

Film Thickness Established by Autoradiography

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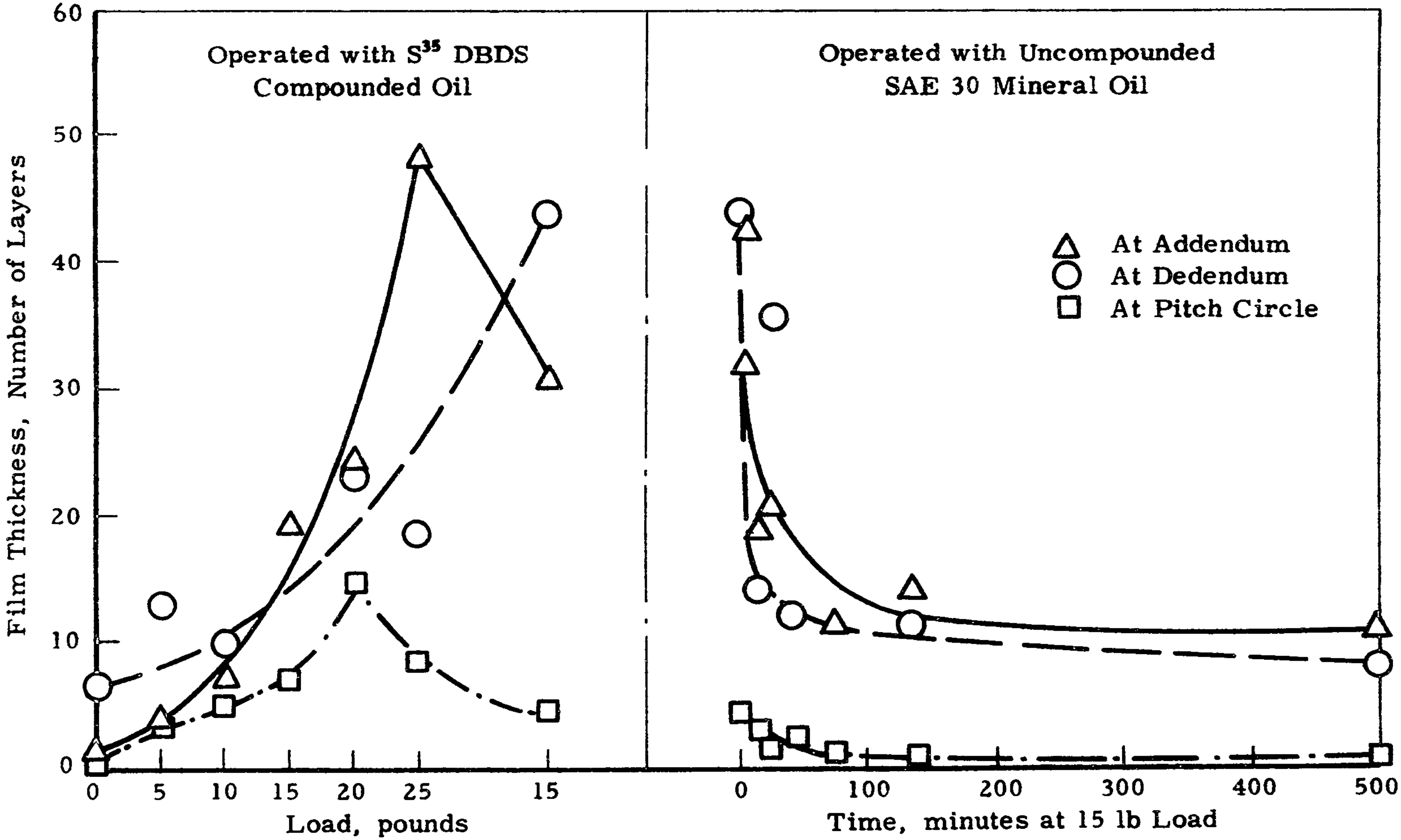

Figure 51. FORMATION AND ATTRITION OF AN EXTREME PRESSURE FILM
Gear - 19 Tooth Fresh Test Gear
Film Thickness Established by Autoradiography

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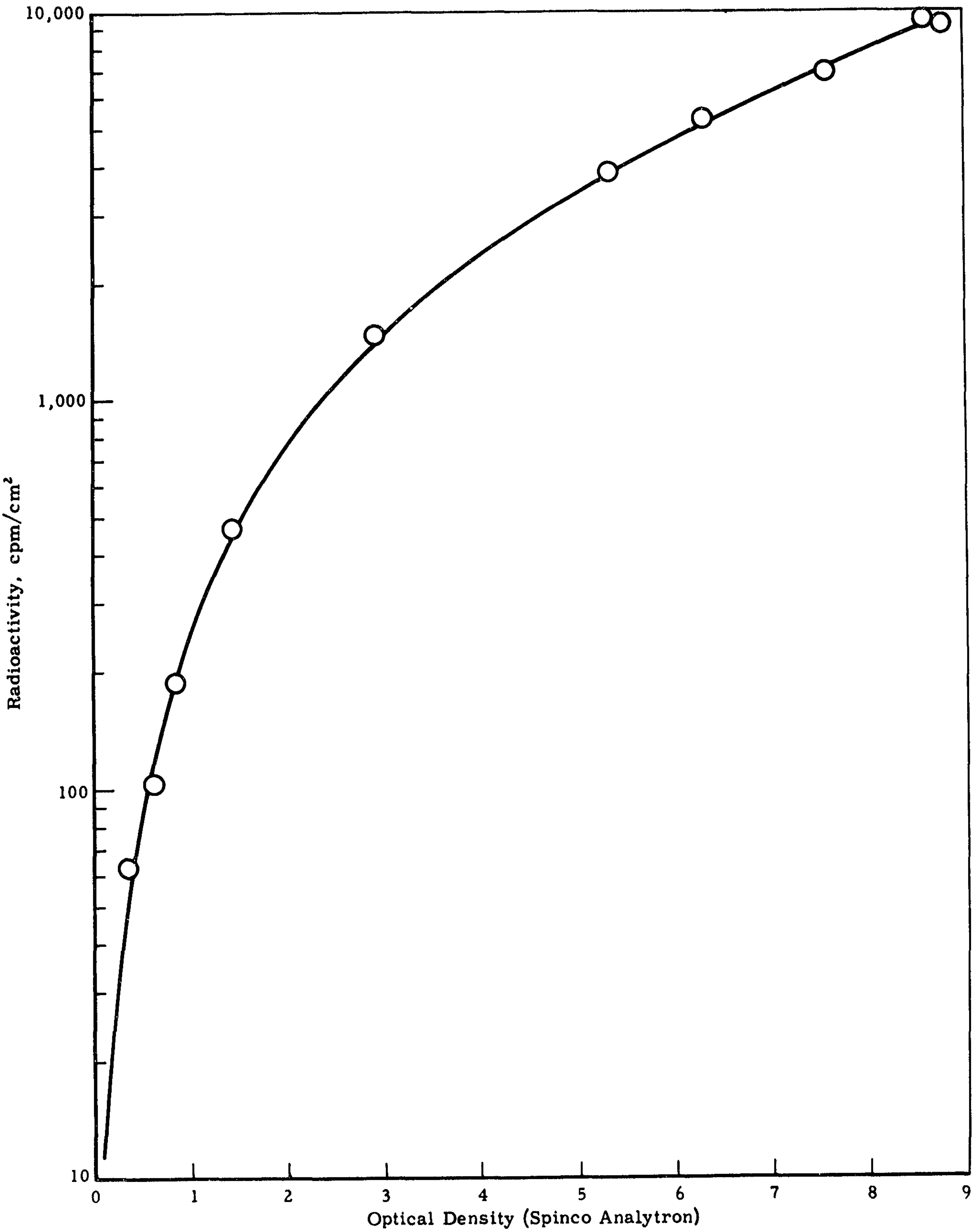

Figure 52. RELATION BETWEEN OPTICAL DENSITY AND RADIOACTIVITY IN cpm/cm^2

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Scoring Performance of E.P. Agents Based on Phosphorus

Discussion of Results. Investigations designed to secure relationships between effective E.P. action and structural configuration and functional groups of additives were continued. Four additional E.P. compounds containing phosphorus were studied. In each case the E.P. additive was dissolved in 1010 grade mineral oil in a concentration of 0.5% by weight of the phorphorus. Tests were conducted in the High Speed Spur Gear Machine at 3200, 5000, 10,000, and 20,000 rpm. The gears were the standard test gears used in this research. The incoming oil temperature was 100°F and the flow rate was 10 cc/sec. Loads were increased in 4 lb beam load steps, running 5 minutes at each load setting. After each run the gears were inspected microscopically. The tests were terminated when the addendum of the pinion gear scored.

The data are presented in Table 20 and in graphical form in Figure 53.

It should be noted that a condition of full scoring was the criterion for the failure of an E.P. compounded oil, as it has been for the evaluations of all E.P. studies in this project. Unlike the scoring phenomena observed for uncompounded oils, in some instances E.P. compounded oils exhibited progressive scoring over variable, and often rather large, load ranges. Such a performance is exemplified by the case of dibutyl phosphite at 3200 rpm where the first evidence of scoring was observed at a 32 lb load, and its reported high rating is based upon the use of complete scoring at 68 lb as the evaluation criterion. The possibility that this arbitrary evaluation scheme may in some instances emphasize unreal differences as well as shade distinctive differences is to be admitted. For the data of these four phosphorus compounds this ambiguity must be considered only in the cases of dibutyl phosphite at 3200 rpm and the three phosphites at 20,000 rpm.

The unequivocal separation of the triphenyl phosphine in virtue of its poor performance is apparent. The comparative evaluation of the other three is not as obvious. Dibutyl phosphite appears to be the best at 3200 and 5000 rpm, but its superiority at 10,000 rpm is not as marked. The data for 20,000 rpm is inconclusive in that conditions of full scoring were not reached.

The pronounced inability of triphenyl phosphine to provide any significant E.P. properties permits some inferences to be made regarding the mechanism of the E.P. action of some phosphorus compounds. As a class, phosphines are very reactive to a number of reagents, reacting vigorously with halogens, forming complex salts with metal derivatives, forming adducts with reagents such as maleic anhydride, etc. They are bases, the degree of basicity being dependent upon the size and type of hydrocarbon moeity. Of particular interest are the various phosphine oxidation products. Primary and secondary phosphines are readily oxidized by atmospheric oxygen, yielding phosphorus and phosphonic and secondary phosphonic acids. On the other hand tertiary phosphines are relatively stable but may be oxidized to the oxides; thus, the hydrolysis or oxidation of triphenyl phosphine to any acid of phosphorus is precluded. It is this property which in large measure chemically distinguishes the separation of triphenyl phosphine from the three phosphites with regards to their E.P. properties. However, it should be noted that since the reaction zone cannot be described accurately, explanations

Table 20. SCORING PERFORMANCE OF FOUR E. P. AGENTS BASED ON PHOSPHORUS

Lubricant	Legend	Speed rpm	Oil Temp °F	Score Load Beam lb	Score Load Tooth lb	Score Load lb/in.	Score Load Hertz Stress psi	HP at Scoring
1010 Mineral Oil	1	3200	100	12	212	852	179×10^3	14.4
		4800	100	4	71	284	102×10^3	7.3
		10000	100	4	71	284	102×10^3	15.0
		20000	100	10	177	710	162×10^3	75.0
1010 Min. Oil + 0.5%w P Triphenylphosphine	2	3200	100	8	141	568	145×10^3	9.8
		5000	100	4	71	284	102×10^3	7.5
		10000	100	8	141	568	145×10^3	30.0
		20000	100	12	212	852	179×10^3	89.9
1010 Min. Oil + 0.5%w P Tri 2-Ethylhexyl-phosphite	3	3200	100	36	636	2556	301×10^3	43.9
		5000	100	16	282	1136	254×10^3	29.9
		10000	100	12	212	852	179×10^3	44.9
		20000	100	40	710	2840	317×10^3	300.
1010 Min. Oil + 0.5%w P Dibutyl Hydrogen phosphite	4	3200	100	68	1200	4828	404×10^3	83.0
		5000	100	48	848	3408	339×10^3	89.8
		10000	100	52	918	3692	353×10^3	195.
		20000	100	32	564	2272	240×10^3	240.
1010 Min. Oil + 0.5%w P Triisopropylphos-phite	5	3200	100	44	776	3124	324×10^3	53.7
		5000	100	12	212	852	175×10^3	22.5
		10000	100	44	776	3124	324×10^3	165.
		20000	100	40	710	2840	317×10^3	300.

Tables 2, 3 and 4 follow

Figure 53 follows

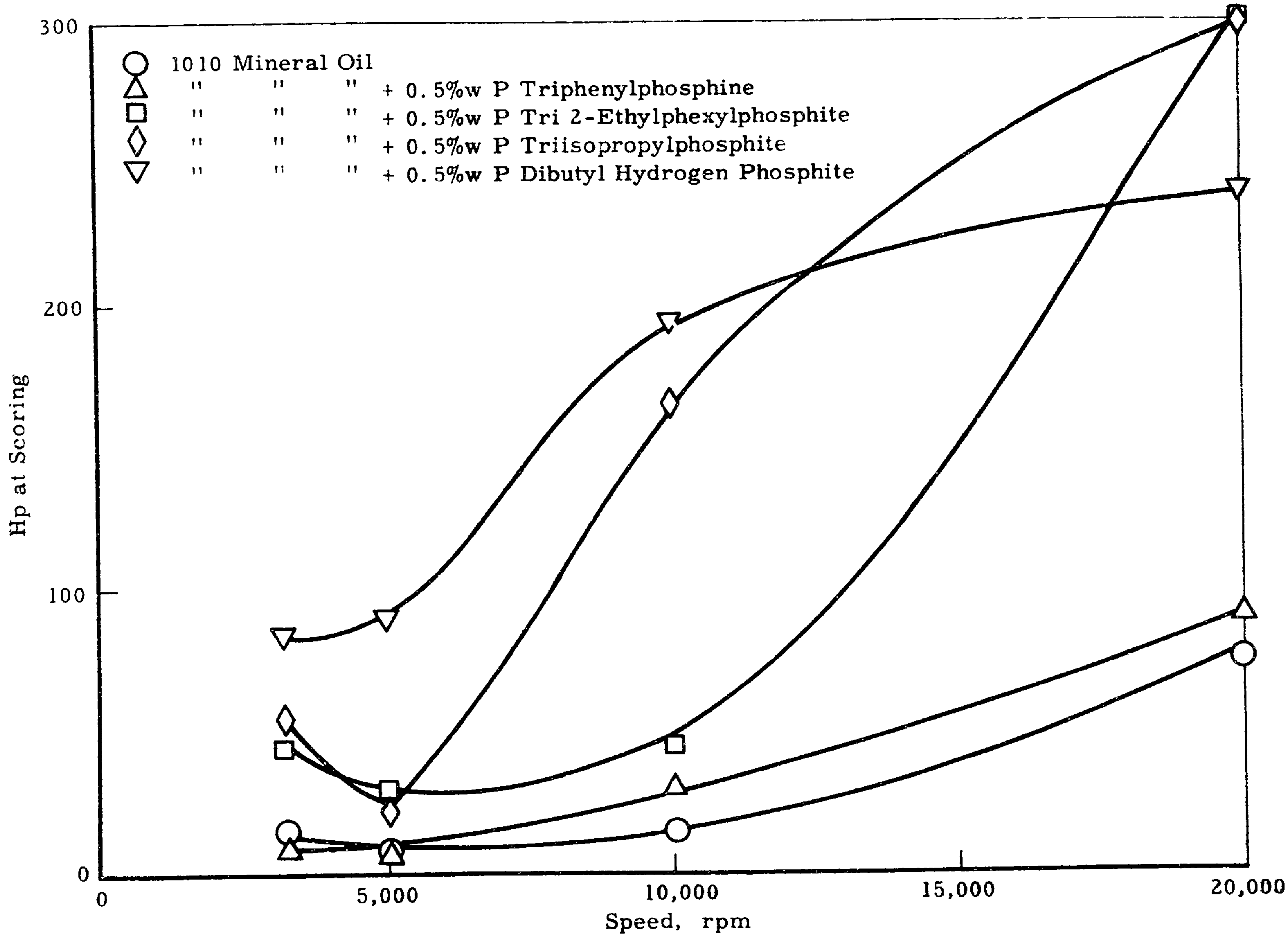

Figure 53. POWER TRANSMITTING CAPACITY VS SPEED FOR E. P. AGENTS BASED ON PHOSPHORUS

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involving other reactions, the susceptibility of these compounds to thermal decomposition, and the reactivities of the various species resulting therefrom are not precluded.

As noted, the secondary phosphite, di-n butylphosphate, is noticeably superior except at 20,000 rpm, where the data are inconclusive. Since secondary phosphites are essentially neutral, this E.P. efficiency does not depend upon a marked increase in acidity compared with the completely esterified phosphites. There are no data to show that for rate or equilibrium considerations the secondary phosphite is favored in the accumulated effects of oxidative, hydrolystic, and thermal decomposition reactions, and deductions by analogy are prevented.

Conclusions. 1. Tertiary phosphines are poor E.P. additives.

2. Di-n butylphosphite is superior to tri-isoproplyphosphite and tri-2-ethylhexylphosphite at 3200, 5000, and 10,000 rpm.

3. There is evidence that E.P. efficiency decreases with increasing size of the alkyl moeity in alkyl phosphites.

Effect of Concentration and Characteristics of Di-Additive Systems

Discussion of Results. A limited study has been made of the effects of concentration in mono-E.P. additive systems and of possible synergism or antagonism in di-E.P. additive systems. The objectives of the concentration variation investigation were to correlate concentrations with E.P. efficiency for the types of additives used and to utilize differences and similarities between the types for further insight into the mechanism of their action. Observations of synergistic or antagonistic effects provide additional data for the construction of reasonable mechanistic models.

In order that distinctive differences of E.P. characteristics would be observed in the concentration study, the selection of the three compounds was based on the prior knowledge that at concentrations containing 0.5% by weight of the specific element, viz. sulfur, phosphorus, or chlorine, considerable E.P. activity was shown by each. In the synergism-antagonism studies two of the pairs were composed of compounds which had shown appreciable E.P. efficiency in mono-additive systems. One component of the third pair had been found to be relatively inert.

The gears, test conditions, and test procedures were as described previously.

The data are given in Tables 21 and 22 and are presented graphically in Figures 54, 55, and 56.

The data for two of the compounds used in the concentration study, viz. triphenylchloromethane and tri n-butylphosphate, demonstrate nothing beyond an abrupt disappearance of E.P. properties when the concentrations were reduced below 0.5% by weight. In the case of the tri n-butyl phosphate the concentration decreases were moderate, being halved for two steps; no E.P. activity was observable at the concentration of 0.25% weight of phosphorus. The concentration of triphenylchloromethane was reduced in two steps by factors of ten. No E.P. activity remained at the 0.05% by weight of chlorine level. The only conclusion to be drawn from these data is that for these two compounds the threshhold concentration for producing observable E.P. action is near 0.5% by weight. This conclusion is not as rigorously applicable for triphenylchloromethane due to the much greater concentration increments.

In contrast with these two systems a system conpounded with elemental sulfur still evidences virtually undiminished activity at the 0.05% by weight level and appreciable activity even at the 0.005% by weight level at the lowest and highest speeds. The data are not available for comparisons with organic sulfur compounds.

The results of the di-E.P. additive systems do not show the same trend of effects at all speeds. At 3200 rpm all E.P. activities were less than that for dibenzyl disulfide alone; the effectiveness of tri-n-butylphosphate was depressed, and dibutyl butyl phosphonate did not lower the activity beyond those of the other two systems. There was no separation of effects at this speed and beyond this leveling the failure of the phosphonate to radically decrease the disulfide E.P. activity was noteworthy.

At 5000 rpm that data reflect no synergism or antagonism. Again the phosphonate failed to depress the disulfide activity.

At 10,000 rpm the E.P. activities of two of the mixtures are apparently due to the disulfide. None of the co-additives reduces the mono-additive value, although each alone carries an appreciably smaller load. On the contrary the disulfide-tri n-butyl phosphate combination is more effective.

At 20,000 rpm the E.P. activity is again apparently due to the dibenzyl disulfide, none of the co-additives reducing the load carrying capacity below that for the disulfide even though in mono additive systems each exhibits much lower activity.

Conclusions. Appreciable E.P. activity is not observable for triphenylchloromethane and tri n-butylphosphate below 0.5% by weight and this may be close to the concentration threshhold for E.P. activity. The E.P. ability of elemental sulfur is unchanged by a ten fold concentration decrease from the 0.5% by weight level at 10,000 and 20,000 rpm; it is lower at the two lower speeds but remains quite large. At the two high speeds the E.P. activity for di-additive systems is that apparently for dibenzyl disulfide The data for 5000 rpm do not support this conclusion as firmly. At 3200 rpm the mixtures show E.P. properties less than those for tri n-butyl phosphate and dibenzyl disulfide alone. The poor E.P. compound dibutyl butyl phosphonate has no greater effect in di-additive systems containing dibenzyl disulfide than do the other two comparatively good E.P. agents.

Table 21. EFFECTS OF CONCENTRATION ON E.P. PERFORMANCE

Lubricants	Speed rpm	Score Load			
		Beam lb	Tooth lb	lb/in.	Hertz Stress psi
1010 mineral oil + 0.5%w triphenylchloromethane	3200	72	1275	5100	424×10^3
	5000	64	1134	4536	392 x "
	10000	60	1080	4320	380 x "
	20000	44	779	3117	325 x "
1010 mineral oil + 0.05%w triphenylchloromethane	3200	8	144	576	139×10^3
	5000	8	144	576	139 x "
	10000	4	72	288	98 x "
	20000	12	216	864	163 x "
1010 mineral oil + 0.005%w triphenylchloromethane	3200	8	144	576	139×10^3
	5000	8	144	576	139 x "
	10000	4	72	288	98 x "
	20000	12	216	864	163 x "
1010 mineral oil + 0.5%w tri-n-butyl phosphate	3200	72	1275	5100	424×10^3
	5000	16	283	1132	196 x "
	10000	12	216	864	163 x "
	20000	20	354	1416	219 x "
1010 mineral oil + 0.25%w tri-n-butyl phosphate	3200	12	216	864	163×10^3
	5000	8	144	576	139 x "
	10000	8	144	576	139 x "
	20000	16	283	1132	196 x "
1010 mineral oil + 0.12%w tri-n-butyl phosphate	3200	8	144	576	139×10^3
	5000	8	144	576	139 x "
	10000	8	144	576	139 x "
	20000	12	216	864	163 x "
1010 mineral oil + 0.5%w sulfur	3200	72	1275	5100	424×10^3
	5000	64	1134	4536	392 x "
	10000	56	991	3965	366 x "
	20000	48	850	3400	339 x "
1010 mineral oil + 0.05%w sulfur	3200	52	920	3680	353×10^3
	5000	36	637	2550	294 x "
	10000	56	991	3965	366 x "
	20000	48	850	3400	339 x "
1010 mineral oil + 0.005%w sulfur	3200	16	283	1132	196×10^3
	5000	8	144	576	139 x "
	10000	8	144	576	139 x "
	20000	24	425	1700	240 x "

Table 22. CHARACTERISTICS OF DI-E.P. ADDITIVE SYSTEMS

Lubricants	Speed rpm	Score Load			
		Beam lb	Tooth lb	lb/in.	Hertz Stress psi
1010 mineral oil + 0.5%w dibenzyl disulfide (DBDS)	3200	52	920	3680	353 x 10³
	5000	20	354	1416	219 x "
	10000	20	354	1416	219 x "
	20000	44	779	3117	325 x "
1010 mineral oil + 0.5%w tri-n-butyl phosphate (TBA)	3200	56	991	3965	366 x 10³
	5000	24	425	1700	240 x "
	10000	12	216	864	163 x "
	20000	32	566	2264	277 x "
1010 mineral oil + 0.5%w dibutyl butyl phosphonate (DBBOA)	3200	8	144	576	139 x 10³
	5000	4	72	288	98 x "
	10000	12	216	864	163 x "
	20000	16	283	1132	196 x "
1010 mineral oil + 0.5%w tricresylphosphate (TCP)	3200	32	566	2264	277 x 10³
	5000	24	425	1700	240 x "
	10000	12	216	864	163 x "
	20000	16	283	1132	196 x "
1010 mineral oil + 0.5%w DBDS + 0.5%w TBA	3200	36	637	2550	294 x 10³
	5000	24	425	1700	240 x "
	10000	32	566	2264	277 x "
	20000	44	779	3117	325 x "
1010 mineral oil + 0.5%w DBDS + 0.5%w DBBOA	3200	40	709	2834	309 x 10³
	5000	20	354	1416	219 x "
	10000	24	425	1700	240 x "
	20000	44	779	3117	325 x "
1010 mineral oil + 0.5%w DBDS + 0.5%w TCP	3200	36	637	2550	294 x 10³
	5000	20	354	1416	219 x "
	10000	20	354	1416	219 x "
	20000	48	850	3400	339 x "

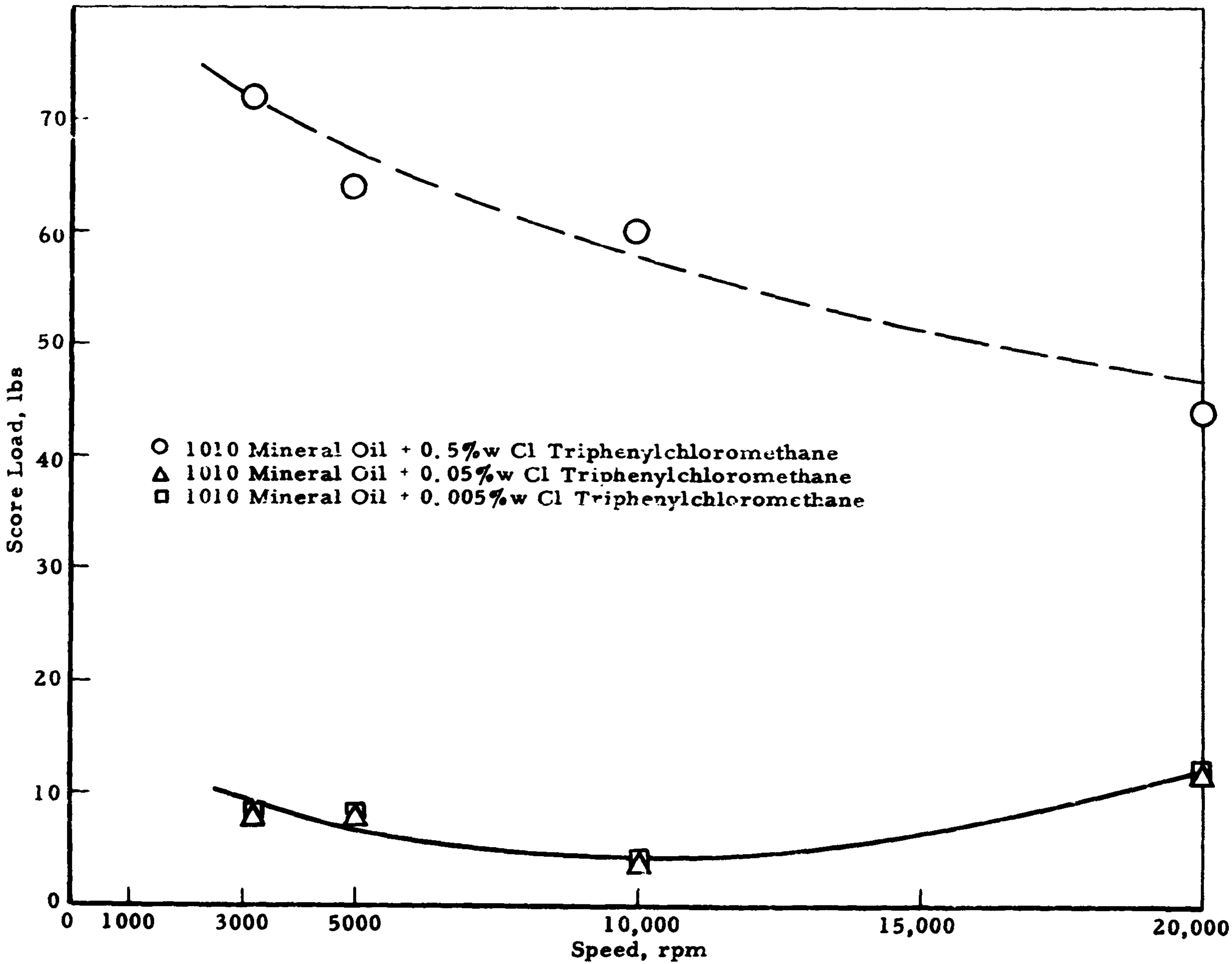

Figure 54. CONCENTRATION EFFECTS WITH TRIPHENYLCHLOROMETHANE

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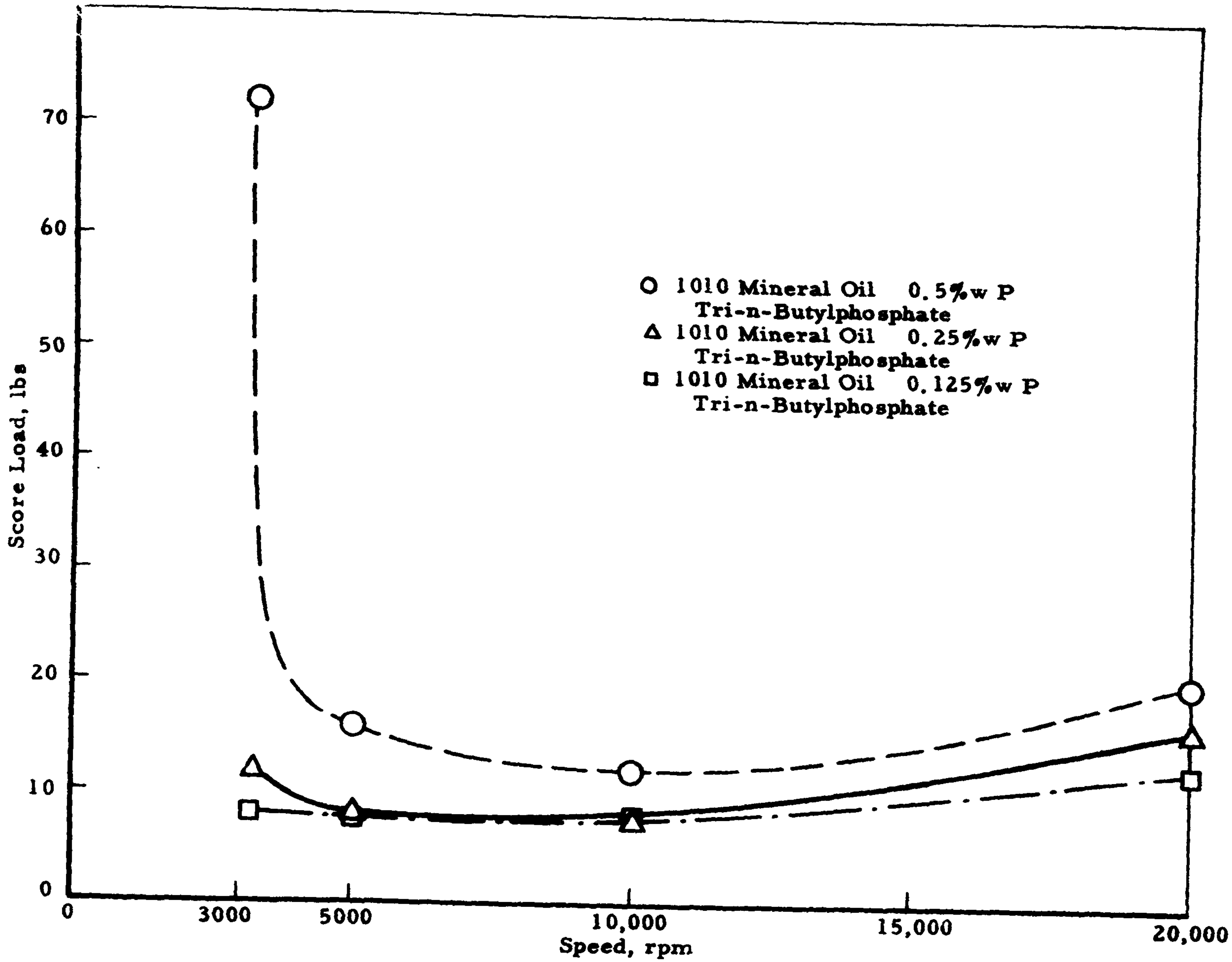

Figure 55. CONCENTRATION EFFECTS WITH TRI-*n*-BUTYLPHOSPHATE

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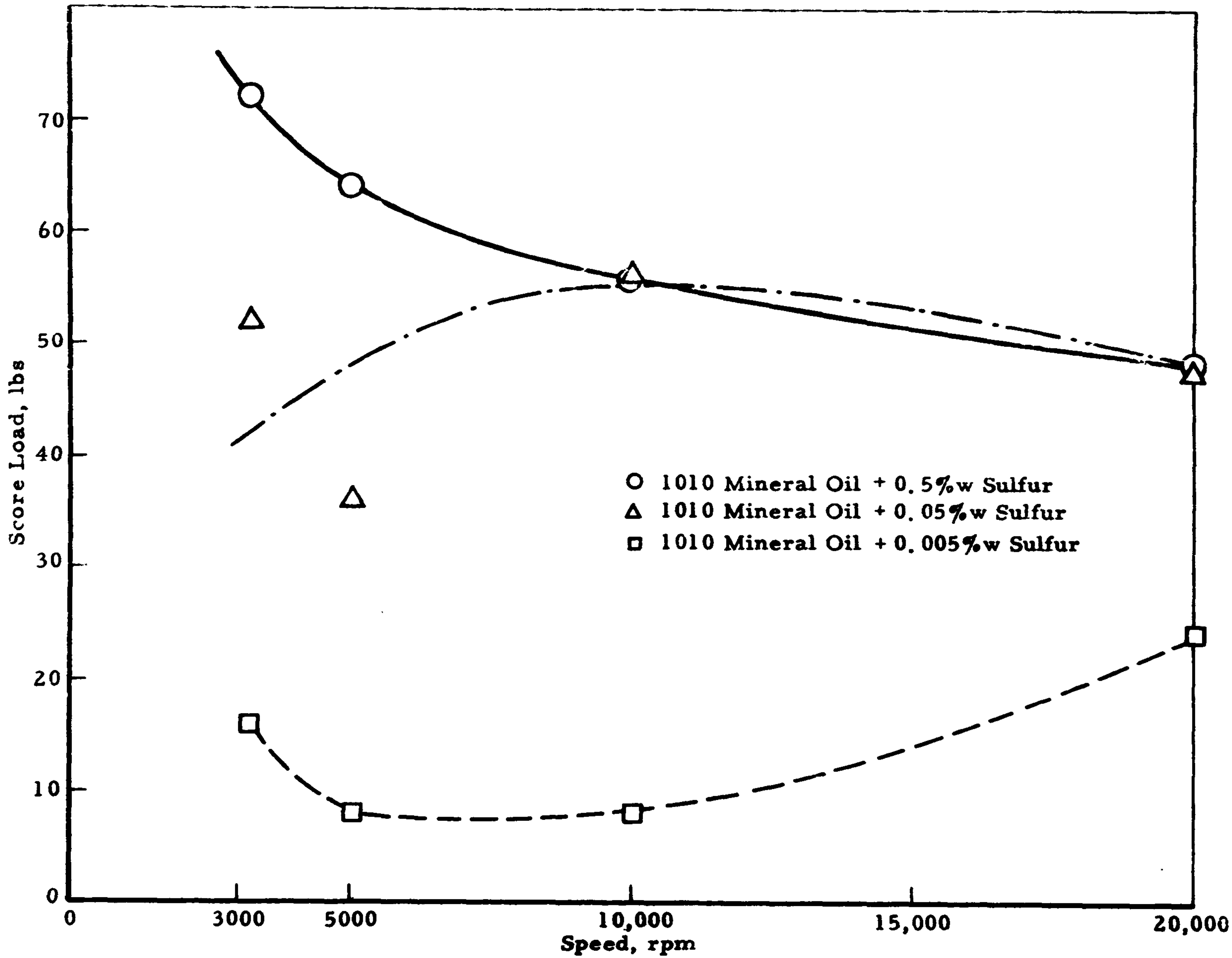

Figure 56. CONCETNRATION EFFECTS WITH SULPHUR

Studies of E.P. Film Formed With Tri n-Butylphosphate

Test Procedure and Results. The use of radioisotopically labelled compounds permits increased sensitivity and accuracy in studies of the kind required in E.P. mechanism investigation, and the necessary quantitative measurements of gear surfaces and of the lubricant are possible in ranges in which other physical measurements fail.

In this study the E.P. compound was tri n-butylphosphate labelled with P^{32}, a phosphorus isotope emitting a 1.7 mev P. The experimental procedures used for dibenzyl disulfide with S^{35} and tri cresyl phosphate with P^{32}, the results of which have been described in reports of this series, were in general followed in this work. In addition, the lubricant was examined further in a series of chemical separation and radiochemical procedures for the purpose of detecting and measuring particular decomposition products of the initial additive.

Discussion of Results. The tri-n-butyl phosphate tagged with P^{32} was purchased from Volk Radiochemical Co. The manufacturer's specifications were 109 mc per 9.4 g as of April 1, 1957. The radioisotopically inert tri-n-butyl phosphate which comprised ca 90% of the mass of the total additive used in the blends had been purified by fractional distillation, the cut at 162°C being used. Two blends were made with SAE 30 mineral oil as the base. The blend used for runs at 3200 rpm contained 854.0 g SAE 30 oil, 33.5 g radioisotopically inactive phosphate, and 4.7 g phosphate with P^{32}. That used for runs at 5300 rpm contained 968.0 g SAE 30 oil, 40.0 g inactive phosphate, and 4.0 g phosphate with P^{32}. If these amounts of components are compared on a %w scale, they result in these numbers for the contents of total phosphate, active phosphate, total phosphorus, and P^{32}, respectively: 4.28 and 4.35, 0.527 and 0.395, 0.51 and 0.52, and 0.063 and 0.048.

The tests were performed at 3200 and 5300 rpm in the slow speed GSL machine. The flow rate of the lubricant was 10 ml per second, and the load increments were 5 lb. Each run at a load setting was 60 minutes long. The gears were the standard 17 and 19 tooth gears.

After each run the gears were removed, washed in a hydrocarbon solvent, monitored, and autoradiographed. The isotope detection procedures made use of a holder for the Geiger tube and film as described and illustrated in the Final Report June, 1954 to June, 1955 of this series. The film was Kodak Personal Monitoring Film, Type 1. The optical densities of the autoradiographs were measured using Spinco Model R Analytrol.

For the purpose of relating optical densities and cpm data complementary measurements were made on steel strips placed in samples of the blends at 100°C and 165°C for various times of immersion. The geometry for counting and autoradiographing was the same as for the gears.

The correlation of cpm per cm^2 with the amount of P^{32} was done by counting samples of the blends which were diluted one to one hundred with toluene and evaporated on a planchet. The relationship of the amount of P^{32} with the thickness of the E.P. film is dependent on the assumptions regarding the nature of the compounds containing the phosphorus in the surface film.

Quantitative measurements of the amounts of the three n-butyl esters of phosphoric acid in the blend used in operations at 30 lb and 3200 rpm were made by a method involving the addition of known amounts of the pure components. The tri-n-butyl phosphate was of the purity described above. The mono- and diesters were prepared by both a controlled alkaline hydrolysis and a separation of a commercial mixture. Having added these pure components, the procedure was to separate the mono- and diesters from the mixture again and determine the radioactivity of samples of each. The monoester was in each case separated from an aqueous ethanol mixture and recovered by a hot water extraction of the solid. The diester was recovered from a hot ethanol extraction of the solid remaining from the complete evaporation of the initial aqueous ethanol solution.

In all instances the cpm for comparable runs in the 3200 rpm series are greater than those for the 5300 rpm series by factors of 2 to 4. Corrections for proportional differences in total or active phosphorus content do not alter these results substantially; corrections based on total phosphate content would increase the gaps by ca 1.6% whereas corrections using the active phosphorus content would decrease the differences by ca 13%. It is evident, thus, that the differences observed are results occasioned by changes in operating conditions, viz., the change in speed. For tri-n-butyl phosphate, then, the amount of phosphorus in the E.P. film is greater by the noted degrees for 3200 rpm as compared with 5300 rpm.

For both speeds the amount of phosphorus is greater on the working side of the gear, compared with the opposite idle side. The data for zero loads show approximately equal values, and it is evident that the condition and the data concur in demonstrating that reactive atmospheres are not extant. The data for the idle side show no marked increases and seem to indicate that temperatures necessary for reactive atmospheres are not created in the vicinities of these faces. On the other hand the increases in the data for the working sides show far greater increases with increasing load severity and inferences can be made of concomitantly increasingly severe reaction conditions as illustrated in Figure 57. No quantitative data which were significant and meaningful were able to be extracted from the autoradiographing procedures. The films for runs at 3200 rpm showed little beyond definite decreases in amount of phosphorus at the pitch circle. In this case complete optical density profiles were prevented by the opaqueness at short distances from the pitch circle. The films for 5300 rpm were within the measurable optical density range and despite some irregularities show definite decreases in phosphorus at the pitch circle. The use of these latter data when correlated via optical density film measurements of phosphorus on the surfaces of steel strips as a result of action by tri-n-butyl phosphate would have yielded estimates of E.P. film thicknesses on the gear teeth by the methods previously demonstrated in this series. Unfortunately, for some inexplicable reasons attempts to secure these data from the steel strip measurements were not successful, and thus the estimates of the depths of the E.P. films are not possible. A typical profile of optical densities is shown in Figure 58.

The examination of the lubricant using radioisotope techniques involved several separations and purifications. No corrections for scattering or self absorption were applied in the counting, since only comparative values are of interest. In the diester determination 0.0035 g were recovered, 1 g of the radioisotopically inactive species having been added. This gave a corrected cpm of 2.2×10^5 based on a 1 g recovery, i.e., a presumed complete recovery of the diester. From data of the relation of the weight of tri-n-butyl phosphate in the oil and cpm it was calculated that the total phosphate present was equivalent to 7.3×10^8 cpm. This count is also of course equivalent, or a measure of, the total phosphorus present. The weight of the diester present, based on a complete recovery of the 1 g of diester added to the system, was obtained by multiplying the original weight of total phosphate added by the ratio of molecular weights of diester to triester and this by the ratio of cpm for the two samples. This weight was calculated to be 8.4×10^{-3} g. This corresponds to a conversion of triester to diester of 0.03%.

Data for the monoester revealed a much smaller quantity present. The procedure of calculation was the same as for the diester. Here a 1 g recovery of the added monoester gave 2.1×10^4 cpm. This corresponds to a conversion of triester to monoester of 0.003% and a weight of monoester of 5.0×10^{-4} g.

It must be noted that although care was used in the separation the possibility of adsorption of the triester on either precipitate is still to be considered. In these calculations this adsorption and also coprecipitation were assumed zero.

Conclusions. The monitoring of the gear teeth surfaces confirms the effect previously noted that the amount of E.P. film increases with increasing load at a specific speed. The data again have demonstrated the relatively large buildup of E.P. film on the working sides of the gear teeth as opposed to the idle sides.

No statements can be made concerning the depth of the E.P. film due to the described unexpected experimental results. However, relative distributions have been shown. The comparative decrease in amount of the film at the pitch circle is clearly evident.

The reasons for the decrease in amount of film at the higher speed are unknown. If it can be assumed to be real and not due to experimental difficulties, it portrays an inverse dependency of amount of film with speed for these two speeds.

The radiochemical analyses of the lubricant have revealed the presence of very small quantities of the mono- and diesters. Remembering that only very small amounts of active species are required for surface reactions and that these amounts are small fractions of what may be present in the lubricant, the results are reasonable. They have, however, only shown the presence of species other than the triester and have not unambiguously pinpointed the actual active species.

Table 23 and Figures 57 and 58 follow

Table 23. FILM FORMATION STUDIES WITH TRI-n-BUTYL PHOSPHATE (P^{32})

Speed	Date	Load, lb	Back-ground, cpm	Decay Corr, %		17 Tooth Gear												19 Tooth Gear											
						Without Shields				Working Side				Opposite Side				Without Shields				Working Side				Opposite Side			
				Based on First Day	Based on Apr. 10	cpm	Corr for BG	Corr for Decay 5-8-57	Corr for Decay 4-10-57	cpm	Corr for BG	Corr for Decay 5-8-57	Corr for Decay 4-10-57	cpm	Corr for BG	Corr for Decay 5-8-57	Corr for Decay 4-10-57	cpm	Corr for BG	Corr for Decay 5-8-57	Corr for Decay 4-10-57	cpm	Corr for BG	Corr for Decay 5-8-57	Corr for Decay 4-10-57	cpm	Corr for BG	Corr for Decay 5-8-57	Corr for Decay 4-10-57
3200 rpm	Apr. 10	0	30	0	0	1674	1644		1044	391	361		361	389	359		359	1312	1282		1282	156	126		126	130	100		100
"	11	5	25	4.5	4.5	2567	2542		2657	720	695		726	433	408		426	2019	1994		2084	753	738		771	391	366		382
"	12	10	39	9.0	9.0	2862	2823		3077	979	940		1024	399	360		382	3574	3535		3853	1157	1118		1218	283	244		290
"	15	15	32	21	21.0	3321	3289		3997	1270	1238		1504	427	395		480	3685	3653		4438	1289	1257		1528	324	292		357
"	16	20	33	24.5	24.5	4197	4164		5204	1566	1533		1309	506	473		589	3989	3956		4926	1619	1586		1976	378	345		430
"	18	25	37	31.5	31.5	4766	4529		5957	1871	1834		2412	607	570		750	4506	4469		5870	1737	1700		2235	427	390		515
"	19	30	36	34.5	34.5	5118	5082		6837	1341	1305		2502	665	629		846	4739	4703		6328	1820	1784		2400	799	763		1027
5300 rpm	May 8	0	32	0	73.0	385	353	353	610	101	69	69	119	87	55	55	95	477	445	445	770	102	70	70	121	101	69	69	119
"	9	5	30	4.5	74.5	657	627	655	1044	275	245	256	427	106	76	79	133	861	831	868	1450	301	271	283	473	86	56	59	98
"	13	10	45	21.0	78.8	740	695	841	1244	322	277	335	495	112	67	81	120	951	906	1096	1621	373	328	396	586	106	61	74	109
"	14	15	32	24.5	79.8	882	850	1034	1529	350	318	396	572	115	83	103	149	1018	986	1228	1772	418	386	481	694	111	79	98	142
"	15	20	33	28.0	80.7	1018	985	1260	1790	444	411	526	743	126	93	119	168	1094	1061	1359	1918	447	414	530	748	112	79	101	145
"	16	25	32	31.5	81.7	1152	1120	1472	2036	540	508	668	923	156	124	163	226	1223	1191	1567	2167	501	469	617	852	112	80	105	145
"	20	30	34	43	84.6	1189	1155	1652	2135	571	537	768	992	140	106	151	196	1253	1219	1747	2251	522	488	698	902	121	87	124	101

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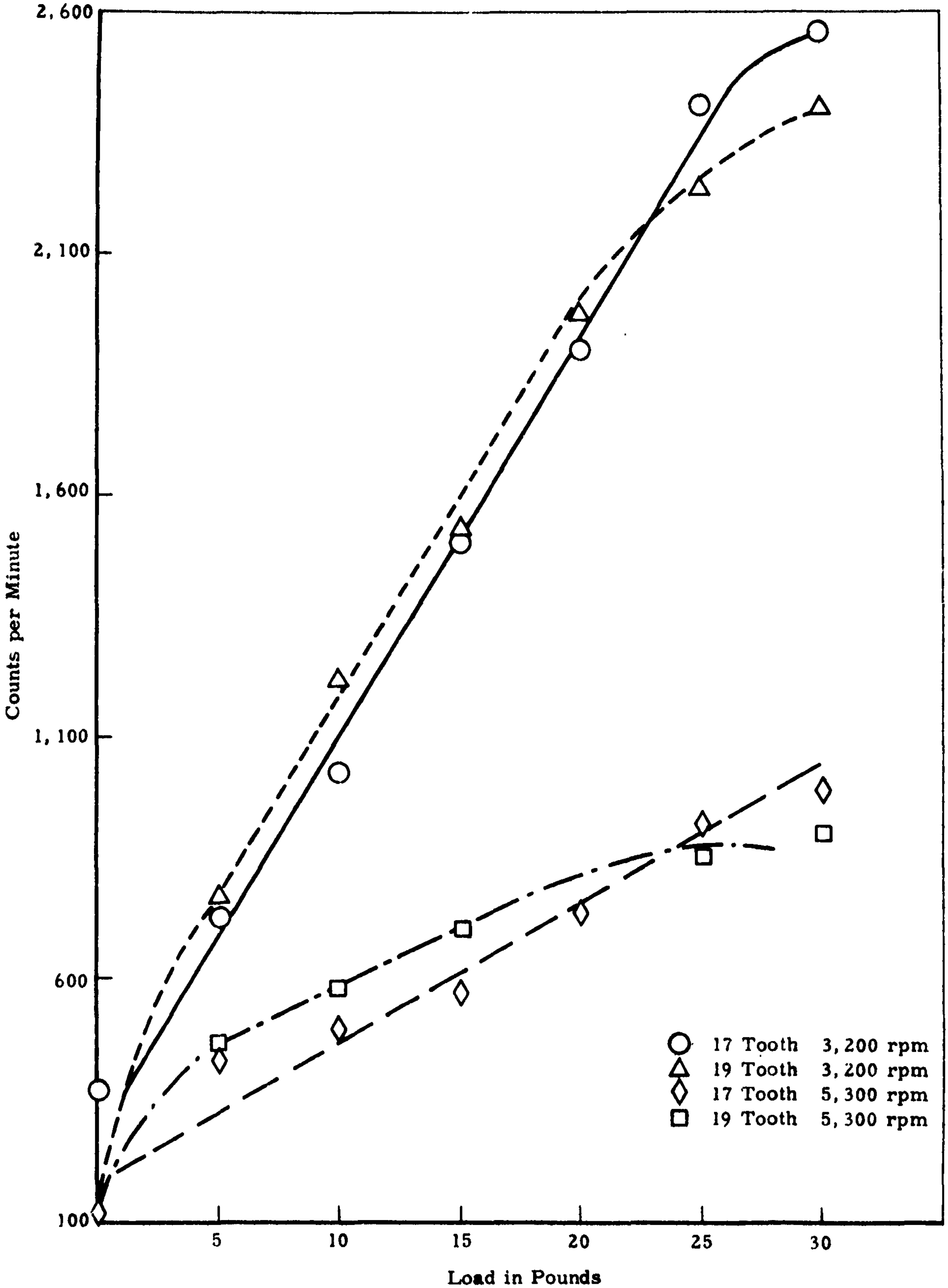

Figure 57. EFFECT OF LOAD ON AMOUNT E. P. FILM

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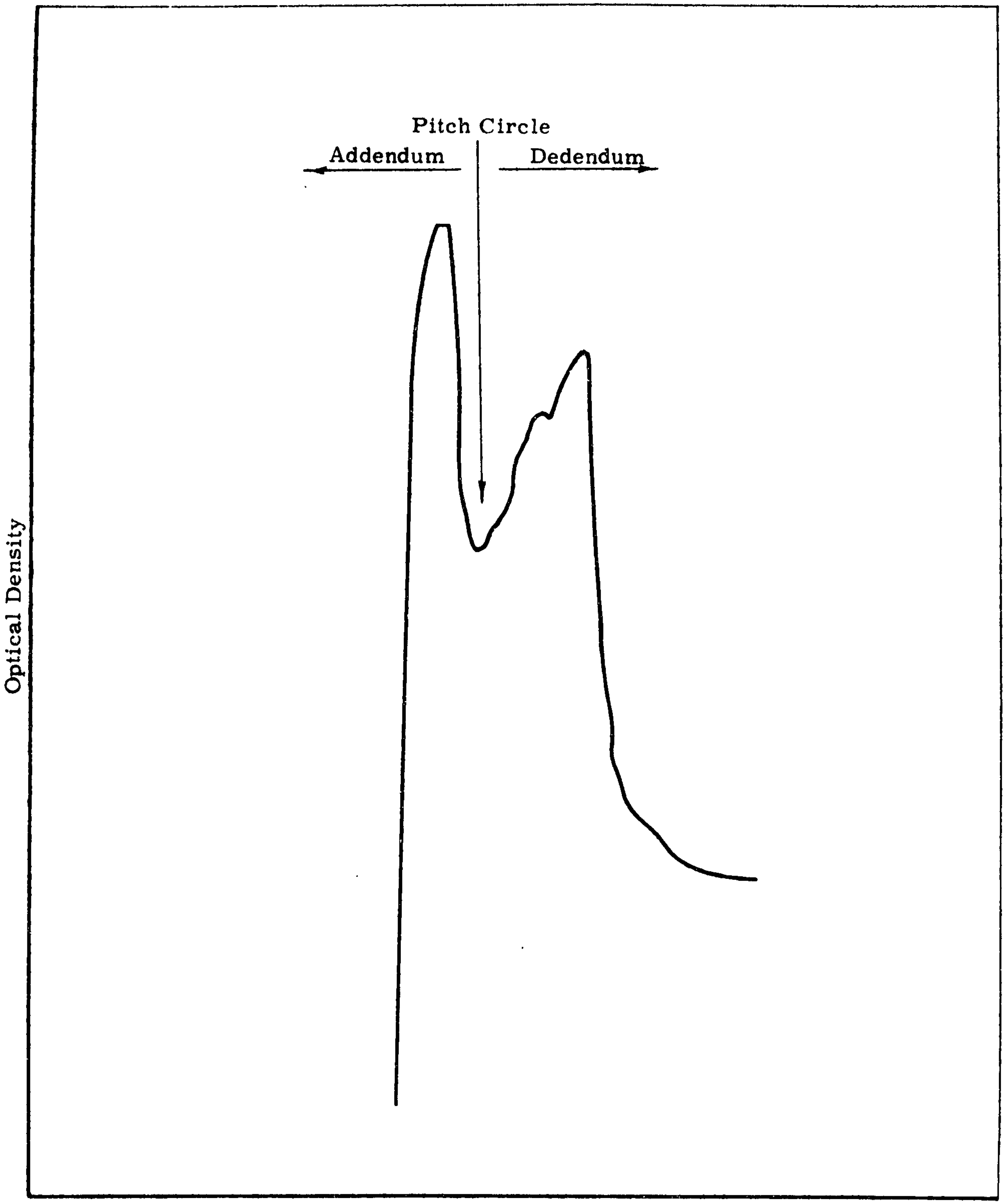

Figure 58. DISTRIBUTION OF E. P. FILM ON GEAR TOOTH

Concluding Remarks

Brief Review of the Work Performed

The purpose of the research on Fundamentals of Gear Lubrication was to secure all possible evidence which could be helpful in explaining the mechanism of gear lubrication. This research was carried out for the past four years and the highlights of the work performed are given below.

First, gear failures were re-examined and were condensed to five basic ones. The choice of scoring as a "limit" for gear performance was explained. Then all known variables affecting gear lubrication were listed and subdivided into three groups: (a) gear geometry and construction factors, (b) operating variables and (c) lubricants properties. The reasons were given for the investigation of some of these variables in preference to others. Third, the test equipment and procedures designed for these investigations were described. The actual experimental work consisted of studying the effects of various gear geometry and construction factors and operating variables. Each of these variables was studied with a large number of different mineral, synthetic and extreme pressure compounded oils so that the effects of lubricant properties were simultaneously investigated.

During the first year of this research the variables investigated were gear speed, lubricant supply temperature, lubricant flow-rate, jet velocity, and location of the point of application of lubricant. The gear geometry factors investigated were diametral pitch and tip relief. As supplementary work wear of gears was studied using a specially developed radioactive gear wear test technique, and some attempts were made to develop a method for the measurement of the relaxation time of oils and to utilize high speed photography for gear lubrication research.

During the second year of these investigations, the effect of various errors in gear construction and the effect of surface finish were investigated. Gear performance under elevated temperatures and high speeds was studied, and the effect of the viscosity of various synthetic oils and the performance of SD-17 lubricant were investigated. Wear studies were expanded by performing a number of long-term wear tests and extreme-pressure lubrication studies were intensified by increasing the number of E.P. agents investigated and studying the mechanism of formation and action of extreme pressure films. These later studies were performed using a specially developed radioactive tracer technique.

In the third year the gears constructed out of different materials were compared for their performances. The data on effect of various errors were expanded, and the effects of temperature and speed on gear wear were studied. During this year the actual studies of viscosity - pressure - temperature characteristics of oils were started, and the studies of extreme pressure lubrication were expanded by investigations of scoring performances of various E.P. agents, studying the nature of chemical reactions between dibenzyl disulfide and iron and expanding the studies of the mechanism of formation, action and attrition of extreme pressure films. In the Summary Report for this year (S-13694) a theory for the mechanism of gear lubrication was formulated.

Finally, in the fourth year of this research the effects of the hardness of gear steel and the width of the gear faces were investigated. The accumulation of the data on the effects of pressure and temperature on viscosities of oils was completed and a series of theoretical studies of the relation between viscosities of lubricants and their load carrying capacities was conducted. Gear operations in the high temperature and high speed ranges were investigated, and the data on the effect of speed on gear wear were expanded. The extreme pressure lubrication studies were expanded by studying scoring performances of various E.P. agents based on phosphorus, studying the effect of concentration and the characteristics of di-additive systems, and expanding the data on formation and attrition of various E.P films.

The results of these investigations permitted the formulation of a theory on the mechanism of gear lubrication. Numerous practical recommendations were also deducted from the results obtained. These are presented below.

Theoretical Conclusions

Two zones of gear lubrication are noted: one in which the meshing gear teeth are separated by a thick lubricating film and the other in which the separating film is of "boundary" or extreme pressure nature. Thick film lubrication may be obtained with either unreactive oils or oils containing extreme pressure agents. The latter perform in the thick film zone only when gears are operated at a load below the score load of their base oil. The boundary or extreme pressure zone of lubrication can be maintained only with oils containing extreme pressure agents.

Thick Film Zone of Lubrication. Rotation of gears in the presence of a lubricant causes the formation of a "hydrodynamic wedge" which tends to separate the meshing teeth with a thick lubricating film, providing the time of contact is sufficiently short and pressure (load) is low enough not to "squeeze" out the lubricant. The evidences obtained during these investigations which confirm the existence of a thick lubricating film are: (1) in operations of gears at loads below the score load of the lubricant used the working surfaces of the teeth preserved their original appearance; (2) wear was undetectable; (3) metal transfer was nonexistent. When the load is gradually increased, a point is finally reached at which scoring occurred. The magnitude of a score load depends on the viscosity and nature of the lubricant, gear geometry and material, and operating conditions. The transition period between the thick film lubrication and scoring with unreactive oils, as a rule, is short and scoring itself is severe and occurs over most of the dedendum and the addendum of the tooth working surface.

The mechanism of scoring is visualized as follows. With an increase in load the pressure within the contact zone is increased, causing a decrease in the thickness of the separating fluid film. Finally, a point is reached when the projecting asperities of contacting surfaces begin to meet. The heat generated by shearing or plastically deforming these asperities causes "thinning" or a decrease of viscosity of the lubricant within the supporting film. As a result, the lubricating film breaks down, and metal to metal contact is established. The heat generated by this "dry" friction is sufficient to melt the surface layers of the metal. A part of the molten metal is stripped off by centrifical forces and a part spreads and adheres to the neighboring areas. Since the actual contact time of each element of the tooth working surface is short and lasts only a small fraction of each revolution, scoring of gears is a very short-time phenomenon. However, it may re-occur again if the severity of the operating conditions is further increased. The instantaneous character of scoring was well demonstrated in our radioactive gear wear studies. Regarding the effects of gear geometry, operating conditions, and lubricant properties on scoring, the findings were that the effect of each factor depends on its ability to assist in the formation and preservation of a thick lubricating film. Speed, however, is an exception since its effects are two-fold. In the slow speed zone (contact time $>10^{-5}$ seconds) score load decreased with increase in speed. In the high speed zone (contact time $<10^{-6}$ seconds) score load increased with increase in speed. The apparent reason for these different effects of speed is the relaxation phenomenon. It is well known that liquids deform according to their viscosities under ordinary velocities, and they respond as an elastic solid when subjected to high deformation rates. In the high speed zone of gear operation the contact time for each element of surface is often shorter than the duration of a deformation pulse causing lubricating oils to act as a plastic solid. The evidences of the peculiar effect of speed were obtained during the first year of these investigations. The subsequent tests, and particularly those for the speeds between 20,000 and 30,000 rpm, confirmed the original findings and conclusions. Regarding the scoring performance of different lubricants, all evidences showed that lubricant viscosity is the most important factor. Thus, various mineral oils, polyglycols and silicone oils showed a consistent relation between their viscosities and load carrying capacities. However, the relation between lubricant viscosity and load carrying capacity does not hold when lubricants of different natures are compared. Originally it was thought that the effect of viscosity, in situ, is the controlling factor and the proofs for correctness of this assumption was sought by establishing the viscosity - pressure - temperature characteristics of the lubricants tested and relating these characteristics to score loads. These studies showed that the differences in performances of various classes of lubricants should be explained in terms of properties other than those of viscosity.

Extreme Pressure Lubrication. Operations of gears with unreactive oils at loads exceeding their load carrying capacity are impractical because the resulting unstable condition leads to the reoccurrence of scoring. However, if extreme pressure agents are added to the unreactive oils, scoring is arrested and operations can be extended to loads often several times higher than the score load of the base oil. The effectiveness of extreme pressure agents to increase the load carrying capacity of unreactive oils is attributed to the protective films formed on the rubbing surfaces by reaction between the extreme pressure agent and the gear metal. The existence of such extreme pressure films on the working surfaces of gear teeth was verified by our experiments with the tagged extreme pressure agents. It was also found that such reactions are highly temperature sensitive and the thickness of the extreme pressure films increased with the severity of operating conditions. For each set of operating conditions an appropriate steady state thickness is established by a continuous formation and attrition of the film. The thinnest film exists on the pitch circle and its thickness increases toward both the addendum and the dedendum. This is in agreement with the disclosed effect of temperature, since generation of heat on the tooth working surface is dependent on the magnitude of sliding, and sliding is zero at the pitch circle and increases rapidly toward both the addendum and the dedendum. The maximum film thickness, as estimated from the data of these investigations, was 1.2 x 10^{-6} inch, which is smaller than the average height of surface irregularities of a surface finished to 20 microinches and therefore lubrication could be classed as being of a "boundary" nature. Generalization of the effect of gear geometry and operating variables in this region of lubrication is difficult, because friction, wear, and scoring behavior of extreme pressure compounded oils depend to a great extent on the nature and concentration of the E.P. agent.

Wear of Gears. Wear of gears operated within the fatigue limits of the metal can be subdivided into three classes: wear by scoring, wear by abrasion and wear by chemical corrosion.

Wear by scoring occurs when the lubricating film between the rubbing surfaces is completely destroyed and dry metal to metal contact is established. If the rates of heat generation by this dry friction are sufficiently high, the upper layers of the metal will be melted and a part of this molten metal will be thrown off the gear by the centrifugal forces. Scoring is a very short time phenomenon and could re-occur only with increasing severity of operating conditions. With unreactive oils, gears predominantly wear by scoring. It follows that load carrying capacity of oils should be inversely related to wear and therefore could be used for predicting relative wear characteristics of unreactive gear oils. This conclusion was verified by the results of these investigations. Scoring can occur also with extreme pressure compounded oils but at higher loads than with the unreactive oils of similar viscosity.

Wear by abrasion is a mechanical process, similar to cutting in a lathe or grinding with an abrasive wheel. In gears it could be caused by extra strong asperities of one surface scratching the other or by an abrasive material present between the meshing teeth. When gears are operated with an unreactive oil at a load below the score load of this oil, wear by abrasion is practically non-existent. With extreme pressure compounded oils operating at load higher than score loads of their base oils, wear by abrasion is often present. The magnitude of this wear depends on the type of extreme pressure compounding agent, on its concentration in the oil, and on the operating conditions.

Wear by chemical corrosion is not a problem with unreactive oils. It is one of the characteristics of extreme pressure lubrication. As was explained above, extreme pressure agents operate by chemically combining with the gear metal and thus forming an extreme pressure protective film. The process of formation and attrition of an extreme pressure film is continuous throughout the operation of gears and is responsible for wear by chemical corrosion. This wear under ordinary operating conditions is usually small.

Practical Considerations

A large amount of the data on the effect of various factors affecting gear performance secured in the course of these investigations permitted not only the formulation of a theory on the mechanism of gear lubrication but also presentation of a number of practical recommendations for gear construction, operation and choice of lubricants. It is realized that some of these recommendations are not new and were known to the industry for sometime. They are included only because sufficient evidence substantiating them was obtained in these investigations. It is realized also that some of the items discussed are too involved and only recommendations in most general terms could be given at the present time. For convenience these practical considerations are subdivided into three groups: (a) those concerning gear manufacturing, (b) those connected with gear operation and (c) those discussing the lubricant factors.

Considerations for Gear Construction. 1. Steel because of its strength and surface durability is a most suitable material for construction of power transmitting spur gears.

2. The harder is the working surface of the gears the better is its durability.

3. Both load carrying capacity and wear characteristics of gears improve with the improvement in surface finish.

4. Accuracy in gear construction is highly important. Accurate gears transmit heavier loads and wear less. The accuracy item should include also the proper alignment of the mating gears and preservation of this alignment throughout gears operation.

5. Tip-relieving of gears improve their scoring performance. However, excessive tip-relieving could give opposite results. The limits for tip-relieving are determined by gear geometry, speed and viscosity of the lubricant used.

6. Gears of larger diametral pitches possess generally better scoring and wear characteristics than gears of smaller diametral pitches. The effect of diametral pitch, however, become less pronounced the higher is the speed or the heavier is the lubricant used.

7. Increases in the face width of gears permit transmission of heavier total loads. However, unit load (in lb/in of face, for example) decreases with the increase in face width.

Considerations for Gear Operation. 1. Beside tooth breakage scoring is the most important gear failure. Lubricants vary widely in their ability to prevent scoring. The load carrying capacity of each lubricant varies with gear geometry and operating condition.

2. Other destructive failures of gears operated within their fatigue limits are abrasion, chemical corrosion (with certain extreme pressure agents) and plastic deformation (for soft gear only).

3. The load carrying capacity of a lubricant cannot be predicted but should be measured. Knowing load carrying capacity of an oil at one speed, load carrying capacity of the same oil at different speeds could be calculated using an empirical equation:

$$\text{Log L.C.C.} = \log \alpha + (n + 1) \log S$$ (See page 11)

This equation is operative for unreactive oils only and for speeds not exceeding 5,000 rpm.

4. Load carrying capacities of unreactive oils are inversely related to wear and therefore can be used as a measure of relative wear characteristics of unreactive lubricants.

5. For economical power transmission it is beneficial to operate the gears at highest possible speeds, because power transmitting capacity of gears increased with the increase in speed.

6. Load carrying capacity of gears decreased with the increase in speed in the slow speed range (contact time more than 10^{-5} sec) and increased in the high speed range (contact time less than 10^{-6} sec).

7. An increase in the temperature of gear operation is accompanied by some decrease in load carrying capacity of oils. However, at temperatures of 400°F or higher most of the lubricants form gummy deposits on gear working surfaces which act as a protective coating and may somewhat improve the scoring performances of gears and gear lubricants.

8. Prolonged operations of gears at temperatures of 600°F or higher with ordinary mineral and synthetic oils are inadvisable because of thermal instability of these lubricants.

9. The amount of lubricant supply to the mating gears should be sufficient and without interruption. Thus, not only a proper flow-rate, but also a proper jet velocity and a correct location of oil supply nozzles are necessary.

Considerations of Lubricants Nature and Properties. 1. Two main classes of gear oils are recognized, the unreactive and the reactive. All uncompounded mineral oils and synthetic oils comprise the class of unreactive lubricants. Some oils compounded (or naturally incorporated) with extreme pressure agents are classed as reactive. Very generally, load carrying capacities of unreactive oils are much lower than of reactive oils of similar viscosity.

2. Scoring performance of unreactive oils depends upon the nature of the lubricants. Thus, comparing the oils of similar viscosity in operations at normal temperatures, silicates and di-esters showed the highest load carrying capacities. They were followed by polyglycols, mineral oils and finally silicones. At elevated temperatures (300°F or higher) silicates were first to lose their effectiveness, and polyglycols followed.

3. The effectiveness of oils of the same nature depends on their viscosity. The higher is the viscosity of an oil the higher is its load carrying capacity.

4. The effectiveness of reactive or extreme pressure lubricants depends on the nature of the additive, the way this agent is incorporated in the oil and its concentration. Viscosity of extreme pressure lubricants is as important as in unreactive oils.

Research Recommendations

The results of these investigations permitted the formulation of a theory of the mechanism of gear lubrication and an explanation of the mechanism of gear wear in operations within the fatigue limits of the gear metal and supplied additional information on the mechanism of action of extreme pressure compounding agents. Based on these results the effects of different factors affecting gear lubrication were explained and a number of recommendations for gear construction and operation were submitted. However, it is realized that the research on Fundamentals of Gear Lubrication, in its entirety, is not completed. Additional investigations with gears of different geometry to clarify the theories and explanations presented are desirable. Items, such as the effect of fatigue on gear performance, "dry" lubrication of gears, and the effect of pre-treatment of surfaces should be investigated. In addition, the phenomena observed during this research and for which the conclusions are tentative should be studied further. For example, the observations that an increase in speed is accompanied by the decrease in load carrying capacities of oils in the speed range between 1000 and 5000 rpm, and that synthetic oils such as di-esters yield higher load carrying capacities than do mineral oils of similar viscosity should be studied further. The work on the operation of gears at temperatures above 400°F should be continued and the relaxation phenomenon with lubricating oils should be more completely investigated.

In broad aspect the mechanisms of E.P. lubrication have been investigated in this research by consideration of the effects of differing structure and functional groups of the additive. Some conclusions although limited in generality, have been reported for additives containing sulfur and chlorine. The data for phosphorus containing compounds have permitted only intercompound comparisons in a narrow range and have not led to deductions concerning E.P. lubrication mechanism.

A continuing, systematic study using this same general approach would allow further deductions of E.P. mechanisms. The research scheme is simple and judicious selections of compounds to be studied would serve to extend, amplify, and reenforce conclusions regarding the requirements for and mechanisms of E.P. action.

There is at least one alternative type of research which can be done in this field. The nature of the E.P. film itself may be investigated. This would entail studies of reactions at high pressures and temperatures and would necessitate qualitative and quantitative analyses of very small amounts of materials. Kinetic and equilibrium data would be necessary. The experimental conditions would be extreme and present techniques would have to be refined and improved to make successful measurements. The type of information secured would provide direct characterizations of E.P. actions and mechanisms.